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Thermodynamic Optimization*

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1. Introduction

Second law analysis in the design of thermal and chemical processes has received considerable attention since 1970s. For example, Gaggioli and Petit (1977) reviewed the first and second laws of thermodynamics as an introduction to an explanation of the thesis that energy analyses of plants, components, and processes should be made by application of the second law that deals with the availability of energy or the potential energy. They illustrated their methodology suggested by applying it to an analysis of the Koppers-Totzek gasification system. Optimization of heat exchangers based on second-law rather than first-law considerations ensures that the most efficient use of available energy is being made.

Second-law analysis has affected the design methodology of different heat and mass transfer systems to minimize the entropy generation rate, and so to maximize system available work. Many researchers considered these processes in terms of one of two entities: exergy (available energy) and irreversibility (entropy production). For instance, McClintock (1951) described irreversibility analysis of heat exchangers, designed to transfer a specified amount of heat between the fluid streams. He gave explicit equations for the local optimum design of fluid passages for either side of a heat exchanger. To the knowledge of authors, McClintock (1951) was the first researcher who employed the irreversibility concept for estimating and minimizing the usable energy wasted in heat exchangers design. Bejan (1977) introduced the concept of designing heat exchangers for specified irreversibility rather than specified amount of heat transferred. Many authors used this technique in the field of cryogenic engineering (Bejan and Smith (1974, 1976), Bejan (1975), and Hilal and Boom (1976)).

One of the first examinations of entropy generation in convective heat transfer was conducted by Bejan (1979) for a number of fundamental applications. Much of the early

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work is well documented in his books (Bejan, 1982a and 1996a). Since the publication of (Bejan, 1996a), entropy generation in internal structure has been examined by numerous researchers. In this section, we will examine these studies that include the optimization of heat exchangers, and enhancement of internal flows. Also, we will proceed to develop some of the basic principles and examine selected results from the published literature.

1.1 Optimization of heat exchangers

In the past thirty five years, much work relating to heat exchanger design based on the second law of thermodynamics was presented by researchers (Bejan, 1988). Heat exchangers have often been subjected to thermodynamic optimization (or entropy generation minimization) in isolation, i.e., removed from the larger installation, which uses them. Examples include the parallel flow, counterflow, crossflow, and phase-change heat exchanger optimizations. We will talk in details about this in this section.

Bejan (1977) presented a heat exchanger design method for fixed or for minimum irreversibility (number of entropy generation units, N_s). The researcher obtained the number of entropy generation units (N_s) by dividing entropy generation rate by the smallest heat capacity rate of the fluids. The value of N_s can range between 0-∞. The heat exchanger would have a better performance if the entropy generation was at its minimum ($N_s \rightarrow 0$). This dimensionless number can clearly express how a heat exchanger performance is close to an ideal heat exchanger in terms of thermal losses. He showed that entropy generation in a heat exchanger is due to heat transfer through temperature gradient and fluid friction. In contrast with traditional design procedures, the amount of heat transferred between streams and the pumping power for every side became outputs of the N_s design approach. Also, he proposed a methodology for designing heat exchangers based on entropy generation minimization. To illustrate the use of his method, the paper developed the design of regenerative heat exchangers with minimum heat transfer surface and with fixed irreversibility N_s.

The thermal design of counterflow heat exchangers for gas-to-gas applications is based on the thermodynamic irreversibility rate or useful power no longer available as a result of heat exchanger frictional pressure drops and stream-to-stream temperature differences. The irreversibility (entropy production) concept establishes a direct relationship between the heat exchanger design parameters and the useful power wasted due to heat exchanger nonideality.

Bejan (1978) demonstrated the use of irreversibility as a criterion for evaluation of the efficiency of a heat exchanger. The researcher minimized the wasted energy using the optimum design of fluid passages in a heat exchanger. He studied the interrelationship between the losses caused by heat transfer across the stream-to-stream due to differences in temperatures and losses caused by fluid friction. He obtained the following relation for the entropy generation rate per unit length as follows:

$$\frac{d\dot{S}_{gen}}{dx} = \frac{\dot{m}}{\rho T}\left(-\frac{dP}{dx}\right) + \frac{dq}{dx}\frac{\Delta T}{T^2\left(1+\frac{\Delta T}{T}\right)} \cong \frac{\dot{m}}{\rho T}\left(-\frac{dP}{dx}\right) + \frac{dq}{dx}\frac{\Delta T}{T^2\left(1+\frac{\Delta T}{T}\right)} \geq 0 \qquad (1)$$

The first term in expression (1) is the entropy production contribution due to fluid friction in the fluid duct. The second term in expression (1) represents the contribution due to heat transfer across the wall-fluid temperature difference. These two contributions were strongly interrelated through the geometric characteristics of the heat exchanger. It should be noted that the use of density (ρ) instead of the inverse of specific volume (v) in the first term on the right hand side. Also, the denominator of the second term on the right hand side was simplified by assuming that the local temperature difference (ΔT) was negligible compared with the local absolute temperature (T). Heat transfer losses could be reduced by increasing the heat transfer area, but in this case pressure drops in the channels increased. Both heat transfer losses and frictional pressure drops in channels determined the irreversibility level of heat exchanger.

A remarkable feature of Eq. (1) and of many like it for other simple devices is that a proposed design change (for instance, making the passage narrower) induces changes of opposite signs in the two terms of the expression. Then, an optimal trade-off exists between the fluid friction irreversibility and the heat transfer irreversibility contributions, an optimal design for which the overall measure of exergy destruction is minimum, while the system continues to serve its specified function. In order to illustrate this trade-off, use the definition of friction factor (f), Stanton number (St), mass flux (G), Reynolds number (Re), and hydraulic diameter (d_h):

$$f = \frac{\rho d_h}{2G^2}\left(-\frac{dP}{dx}\right) \tag{2}$$

$$St = \frac{dq}{dx}\frac{1}{p\Delta T c_p G} \tag{3}$$

$$G = \frac{\dot{m}}{A} \tag{4}$$

$$\mathrm{Re} = \frac{G d_h}{\mu} \tag{5}$$

$$d_h = \frac{4A}{p} \tag{6}$$

In Eq. (3), the quantity *(dq/dx)/(pΔT)* is better known as the average heat transfer coefficient. The entropy generation rate, Eq. (1) becomes

$$\frac{d\dot{S}_{gen}}{dx} = \left(\frac{dq}{dx}\right)^2 \frac{d_h}{4T^2 \dot{m} c_p St} + \frac{2\dot{m}^3 f}{\rho^2 T d_h A^2} \tag{7}$$

Where heat transfer rate per unit length and mass flow rate are fixed. The geometric configuration of the exchanger passage has two degrees of freedom, the perimeter (p) and the cross-sectional area (A), or any other pair of independent parameters, like (Re; d_h) or (G;

d_h). If the passage is a straight pipe with circular cross-section, p and A are related through the pipe inner diameter d that is the only degree of freedom left in the design process. Writing

$$d_h = d, \quad A = \pi d^2 / 4, \quad and \quad p = \pi d \tag{8}$$

Equation (7) becomes

$$\frac{d\dot{S}_{gen}}{dx} = \left(\frac{dq}{dx}\right)^2 \frac{d_h}{\pi T^2 k Nu} + \frac{32\dot{m}^3 f}{\pi^2 \rho^2 T d^5} \tag{9}$$

Where $Re = 4\dot{m}/\pi\mu d$. The Nusselt number (Nu) definition, and the relation between Nu, St, Re, and the Prandtl number ($Pr = \nu/\alpha$)

$$Nu = \frac{h_{av} d_h}{k} = St.\mathrm{Re}.\mathrm{Pr} = St..Pe \tag{10}$$

Introducing two classical correlations for fully developed turbulent pipe flow (Bejan, 1993),

$$Nu = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \quad (0.7 \langle \ \mathrm{Pr} \ \langle \ 160 : \mathrm{Re} \ \rangle \ 10^4) \tag{11}$$

$$f = 0.046\ \mathrm{Re}^{-0.2} \quad (10^4 \langle \ \mathrm{Re} \ \langle \ 10^6) \tag{12}$$

and combining them with Eq. (9), yields an expression for exergy destruction, which depends only on Re. Differentiating the exergy destruction with respect to the Reynolds number (Re) and equaling the result with zero, we find that the entropy generation rate is minimized when the Reynolds number (or pipe diameter) reaches the optimal value (Bejan, 1982a)

$$\mathrm{Re}_{\mathrm{opt}} = 2.023\mathrm{Pr}^{-0.071}\mathrm{B}^{0.358} \tag{13}$$

Equation (13) shows how to select the optimal pipe size for minimal irreversibility. Parameter B is a heat and fluid flow ''duty'' parameter that accounts for the constraints of heat transfer rate per unit length, and mass flow rate:

$$B = \dot{m}\left(\frac{dq}{dx}\right)\frac{p}{\mu^{5/2}(kT)^{1/2}} \tag{14}$$

Additional results may be obtained for non-circular ducts using the appropriate expressions for the geometry A and p, and appropriate models for heat transfer and friction coefficients.

The Reynolds number (Re) effect on the exergy destruction can be expressed in relative terms as

$$\frac{d\dot{S}_{gen}/dx}{(d\dot{S}_{gen}/dx)_{\min}} = 0.856\left(\frac{\mathrm{Re}}{\mathrm{Re}_{opt}}\right)^{-0.8} + 0.144\left(\frac{\mathrm{Re}}{\mathrm{Re}_{opt}}\right)^{4.8} \tag{15}$$

where the ratio on the left-hand side is known as the entropy generation number (N_s), (Bejan, 1982a). In the denominator of the left hand side of Eq. (15), the minimum exergy destruction is calculated at the optimum Reynolds number (Re_{opt}). Also, $Re/Re_{opt} = d_{opt}/d$ because the mass flow rate is fixed. Using Eq. (15), it is clear that the rate of entropy generation increases sharply on either side of the optimum. The left hand side of the optimum represents the region in which the overall entropy generation rate is dominated by heat transfer effects. The right hand side of the optimum represents the region in which the overall entropy generation rate is dominated by fluid friction effects. The left hand side of Eq. (15) is used to monitor the approach of any design relative to the best design that can be conceived subject to the same constraints. Bejan (1982a, 1988) used this performance criterion extensively in the engineering literature. Also, Mironova et al. (1994) recognized this performance criterion in the physics literature.

Bejan (1978) also made a proposal to use the number of entropy production units (N_s) as a basic yardstick in describing the heat exchanger performance. This dimensionless number was defined as the entropy production rate or irreversibility rate present in a heat exchanger channel. When $N_s \to 0$, this implied an almost ideal heat exchanger channel. According to his study, it was enough to increase the effectiveness by using design criterions like the minimization of difference wall temperature or maximization of the ratio of heat transfer coefficient to fluid pumping power.

Bejan (1979) illustrated the second law aspects of heat transfer by forced convection in terms of four fundamental flow configurations: pipe flow, boundary layer over flat plate, single cylinder in cross-flow, and flow in the entrance region of a flat rectangular duct. The researcher analyzed in detail the interplay between irreversibility due to heat transfer along finite temperature gradients and, on the other hand, irreversibility due to viscous effects. He presented the spatial distribution of irreversibility, entropy generation profiles or maps, and those flow features acting as strong sources of irreversibility. He showed how the flow geometric parameters might be selected to minimize the irreversibility associated with a specific convective heat transfer process.

Bejan (1980) used the second law of thermodynamics as a basis for evaluating the irreversibility (entropy generation) associated with simple heat transfer processes. In the first part of his paper, he analyzed the irreversibility production from the local level, at one point in a convective heat transfer arrangement. In the second part of his paper, he devoted to a limited review of second law analysis applied to classic engineering components for heat exchange. In this category, the paper included topics like heat transfer augmentation techniques, heat exchanger design, and thermal insulation systems. The researcher presented analytical methods for evaluating and minimizing the irreversibility associated with textbook-type components of heat transfer equipment. Also, he obtained an expression for the entropy generation rate in a balanced counterflow heat exchanger with zero pressure drop irreversibility as follows:

$$N_s = \ln \frac{\left(1 + \frac{T_1}{T_2} NTU\right)\left(1 + \frac{T_2}{T_1} NTU\right)}{(1 + NTU)^2} \tag{16}$$

Using Eq. (16), $N_s = 0$ at both $\varepsilon = 0$ (or at $NTU = 0$) and $\varepsilon = 1$ (or at $NTU = \infty$), and had its maximum value at $\varepsilon = 0.5$ (or at $NTU = 1$). The maximum N_s increases as soon as T_1/T_2 goes above or below 1:

$$N_{s,\max} = \ln\left[\frac{1}{2} + \frac{1}{4}\left(\frac{T_1}{T_2} + \frac{T_2}{T_1}\right)\right] \tag{17}$$

N_s increases with the absolute temperature ratio T_2/T_1. When $N_s > 1$, the irreversibility decreases sharply as $\varepsilon \to 1$. On the left side of the maximum $N_s < 1$, the irreversibility decreases due to insufficient heat transfer across a temperature difference of order (T_1-T_2).

This maximum entropy paradox constitutes an excellent illustration of the importance of the principle of thermodynamic isolation in the optimization of an engineering component.

Chowdhury and Sarangi (1980, 1983) used irreversibility analysis to predict the optimum thermal conductivity of the separating wall in a concentric tube counterflow heat exchanger. The researchers accounted for the entropy generation due to axial conduction in the wall, along with that due to lateral heat transfer and fluid friction. The frictional entropy generation was independent of the thermal conductivity of the wall and also did not affect the thermal effectiveness of the heat exchanger. As a result, they treated it as constant throughout this work. They assumed that the entropy generations due to lateral and axial heat transfer were independent of each other.

Chowdhury and Sarangi (1982) studied the generation of entropy in a counterflow heat exchanger. For nearly ideal heat exchanger with nearly balanced capacity rate, the researchers obtained an expression for the number of entropy generation units, N_s. They compared the results of their expression with exact calculation and results of Bejan (1977). They observed that their new expression gave a much closer approximation and also could be easily incorporated into the new design procedure of Bejan.

Bejan (1982a) showed that the Entropy Generation Minimization (*EGM*) method was dependent on the use of fluid mechanics, heat transfer, and thermodynamics in its application. The difference between the exergy method and the entropy generation minimization method is that exergy method uses only the first law, second law, and the properties of the environment. On the other hand, *EGM* characteristics are system modeling, development of the entropy generation rate as a function of the model parameters and the ability to minimize the entropy generation rate.

The researcher applied the entropy generation balance or entropy imbalance equation to a control volume of an open system. For gas-gas heat exchanger, he explained entropy generation as the sum of the entropy generation caused by finite temperature difference with frictional pressure drop.

$$\dot{S}_{gen} = \dot{S}_{gen,\Delta T} + \dot{S}_{gen,\Delta P} \tag{18}$$

The first term on the right-hand side of Eq. (18) is the entropy generation rate accounting for the heat transfer irreversibility, and the second term for the fluid friction irreversibility. He expressed that entropy generation (S_{gen}) = 0 corresponded to the highest quality while the entropy generation (S_{gen}) > 0 represented poorer quality.

Also, he described the relative importance of the two irreversibility mechanisms using the irreversibility distribution ratio (ϕ) that was defined as:

$$\phi = \frac{\text{fluid - flow irreversibility}}{\text{heat transfer irreversibility}} = \frac{\dot{S}_{gen,\Delta P}}{\dot{S}_{gen,\Delta T}} \tag{19}$$

For example, the irreversibility distribution ratio (ϕ) varies along with the V-shaped curve of entropy generation number (N_s), or relative entropy generation rate in a smooth pipe with heat transfer (Bejan, 1980), increasing in the direction of large Reynolds numbers (small pipe diameters because the mass flow rate is fixed) in which the overall entropy generation rate is dominated by fluid friction effects. At the optimum (corresponding to N_s = 1), the irreversibility distribution ratio (ϕ) assumes the value ϕ_{opt} = 0.168. This means that the optimal trade-off between the irreversibility due to heat transfer effects and the irreversibility due to fluid friction effects does not coincide with the design where the irreversibility mechanisms are in perfect balance, even though setting $\phi = 1$ is a fairly good way of locating the optimum.

Substituting Eq. (19) into Eq. (18) yields

$$\dot{S}_{gen} = (1+\phi)\dot{S}_{gen,\Delta T} \tag{20}$$

In addition, augmentation entropy generation number ($N_{s,a}$) was given by

$$N_{s,a} = \frac{\dot{S}_{gen,a}}{\dot{S}_{gen,o}} \tag{21}$$

This definition represents the ratio of the augmented to base channel entropy generation rates. Under particular flow conditions and/or constraints, $N_{s,a} < 1$ is desirable, as the augmented system is thermodynamically improved over the basic system, because, in addition to enhancing heat transfer, the irreversibility degree of the apparatus is reduced assuming other factors like heat transfer duty, pressure drop, or pumping power remain the same. If the function of the heat exchanger passage is fixed (i.e. mass flow rate and heat flux are given), this dimensionless number can be written in the more explicit form (Bejan, 1988)

$$N_{s,a} = \frac{1}{1+\phi_0} N_{s,\Delta T} + \frac{\phi_0}{1+\phi_0} N_{s,\Delta P} \tag{22}$$

In Eq. (22), ϕ_0 represents the irreversibility distribution ratio of the reference design, whereas $N_{s,\Delta T}$ and $N_{s,\Delta P}$ are the values of $N_{s,a}$ in the limits of pure heat transfer irreversibility and pure fluid-flow irreversibility:

$$N_{s,\Delta T} = \frac{St_0 d_{h,a}}{St_a d_{h,0}} \tag{23}$$

$$N_{s,\Delta P} = \frac{f_a d_{h,0} A_0^2}{f_0 d_{h,a} A_a^2} \tag{24}$$

The geometric parameters (A, d_h) before and after augmentation are linked through the constant mass flow rate constraint that reads

$$\mathrm{Re}_a \frac{A_a}{d_{h,a}} = \mathrm{Re}_0 \frac{A_0}{d_{h,0}} \tag{25}$$

Substituting Eq. (23) and Eq. (24) into Eq. (22) yields

$$N_{s,a} = \frac{1}{1+\phi_0}\frac{St_0 d_{h,a}}{St_a d_{h,0}} + \frac{\phi_0}{1+\phi_0}\frac{f_a d_{h,0} A_0^2}{f_0 d_{h,a} A_a^2} \tag{26}$$

Equation (26) shows that $N_{s,a}$ is, in general, a function of both the heat transfer coefficient ratio (St_a/St_0) and the friction factor ratio (f_a/f_0). The numerical value of ϕ_0 dictates the relative importance of the friction factor ratio (f_a/f_0). ϕ_0 is known because the reference design is known. It should be noted that ϕ_0 describes the thermodynamic regime of operation of the heat exchanger passage (ΔT losses versus ΔP losses), much in the way that Re_0 indicates the fluid mechanics regime (laminar versus turbulent).

For the case of no change in hydraulic diameter and the cross-sectional appreciably ($d_{h,a} \cong d_{h,0}$, $A_a \cong A_0$), the augmentation entropy generation number ($N_{s,a}$) has this simple form

$$N_{s,a} = \frac{1}{1+\phi_0}\frac{St_0}{St_a} + \frac{\phi_0}{1+\phi_0}\frac{f_a}{f_0} \tag{27}$$

Bejan (1982b) summarized an important contemporary trend in the field of heat transfer and thermal design. The researcher represented this trend using the infusion of the second law of thermodynamics and its design-related concept of entropy generation minimization. This new trend was important and, at the same time, necessary, if the heat transfer community was to contribute to a viable engineering solution to the energy problem. The examples considered in his article ranged from the irreversibility associated with some of the most fundamental convective heat transfer processes, to the minimum irreversibility design of one-dimensional insulations like the main counterflow heat exchanger of a helium liquefaction plant.

Bejan (1983) discussed the irreversibility characteristics of the heat exchangers in which at least one of the streams was a two-phase mixture.

Witte and Shamsundar (1983) defined a thermodynamic efficiency based on the second law of thermodynamics for heat exchange devices. The efficiency could be simply written in terms of the mean absolute temperatures of the two fluids exchanging heat, and the appropriate environment temperature, Their expression was

$$\eta_{W-S} = 1 - \frac{T_o \dot{S}_{gen}}{\dot{Q}} \tag{28}$$

$$\dot{Q} = \dot{m}_h(h_{in} - h_{out})_h = \dot{m}_c(h_{out} - h_{in})_c \tag{29}$$

$\eta_{W\text{-}S}$ = 1 represented the highest value and corresponded to the reversible process. The examination of this efficiency indicated that $\eta_{W\text{-}S}$ could be negative, and the full range of this efficiency was $-\infty < \eta_{W\text{-}S} \leq 1$. Negative values of $\eta_{W\text{-}S}$ characterized counterflow heat exchangers working at cryogenic operational conditions. It should be noticed that this is a conceptually inconvenient result ((Bejan, 1988), (Hesselgreaves, 2000)).

Also, Witte and Shamsundar (1983) showed that for a given ratio of hot to cold inlet temperatures, the efficiency and effectiveness for particular heat exchange configurations were related. They compared this efficiency to second-law efficiencies proposed by other authors, and showed to be superior in its ability to predict the influence of heat exchanger parameter changes upon the efficiency of energy use. They applied this concept to typical heat exchange cases to demonstrate its usefulness and sensitivity.

London and Shah (1983) presented an operationally convenient methodology for relating economic costs to entropy generation. This methodology, in the hands of the heat exchanger designer, allowed an interaction with the system designer to gain insights into the trade-offs allowed between the thermodynamic irreversibilities of flow friction, heat transfer, heat leakage, and mixing. This methodology started with recognition of the appropriate individual irreversibilities. Then, it related the individual costs to system rating and energy penalties by thermodynamic arguments. The analysis loop was closed by considerations related to reduction of the individual irreversibilities in a cost-effective way. On the other hand, the usual energy or "exergy" analysis provided an answer for the overall costs of the collective irreversibilities. This did not provide the engineer with the insight needed to minimize the individual irreversibilities in a cost-effective manner.

Perez-Blanco (1984) discussed irreversibility in heat transfer enhancement. The researcher developed the methods of calculating overall entropy generation rate in a single-flow heat exchanger tube with uniform wall temperature.

Sekulic and Baclic (1984) considered the concept of enthalpy exchange irreversibility (*EEI*). The researchers conducted the optimization of heat exchangers on the basis of entropy generation number for counterflow and crossflow heat exchangers.

da Costa and Saboya (1985) discussed second law analysis for parallel flow and counterflow heat exchangers. In a comparative study of the irreversibility due to heat transfer for imbalanced (i.e. the thermal capacity rates for both fluids are not the same) counterflow and parallel flow heat exchangers, the researchers found that the maximum occurs at effectiveness (ε) = 1 in parallel flow heat exchangers.

Sekulic (1985-1986) presented a note on the thermodynamic approach to the analysis of unequally sized passes in two-pass crossflow heat exchangers.

Sekulic and Herman (1986) considered the minimum of enthalpy exchange irreversibility (*EEI*) as a selective criterion in heat exchanger design. The researchers applied this concept in the core sizing procedure of a compact crossflow heat exchanger for gas-to-gas application. In the final analysis, the approach objective was the pressure drop choice in such a way that from the total set of possible heat exchanger core dimensions the thermodynamically optimal one was selected.

Sekulic (1986) applied the entropy generation (irreversibility) concept founded on the second law of thermodynamics in heat exchanger analysis. In this analysis, the quantity termed enthalpy exchange irreversibility norm (*EEIN*) was the measure of the internal heat exchanger irreversibilities. The researcher discussed the behavior of *EEIN* as a function of the heat exchanger thermal size for an arbitrary flow arrangement and more precisely for two characteristic limiting cases: cocurrent and countercurrent heat exchangers.

In the heat exchangers design, the enhancement of heat transfer surface area is effective to reduce the loss due to the fluid-to-fluid temperature difference. On the other hand, this leads to the increase of the pressure loss in the channel. The optimum working condition must be determined by taking these conditions trade into account. As a result, Tsujikawa et al. (1986) presented the design method of the regenerator of the gas turbine cycle applied with the entropy generation from the viewpoint of the second law of thermodynamics. Their study was mainly concerned with the optimization through the choice of the minimum entropy production. For the fixed value of the pressure ratio of the compressor, the researchers calculated the number of entropy generation units and determined the optimum temperature efficiency of the generator that gave the minimum heat transfer surface area.

Krane (1987) applied second law analysis techniques based on the minimization of entropy generation to the optimal design and operation of a sensible heat thermal energy storage system in which the storage element was both heated and cooled by flowing streams of gases. His results showed that (1) an entire operational cycle that consisted of a storage process and a removal process must be considered (as opposed to the storage process alone) to optimize the design and performance of such a system; and (2) a typical optimum system destroyed approximately 70-90% of the entering availability and, therefore, had an extremely low thermodynamic efficiency.

Zubair et al. (1987) presented a closed-form analytical method for the second-law-based thermoeconomic optimization of two-phase heat exchangers used as condensers or evaporators. Due to finite temperature difference heat transfer and pressure drops, the researchers proposed the concept of "internal economy" as a means of estimating the economic value of entropy generated, thus permitting the engineer to trade the cost of entropy generation in the heat exchanger against its capital expenditure. They presented results in terms of the optimum heat exchanger area as a function of the exit/inlet temperature ratio of the coolant, unit cost of energy dissipated, and the optimum overall heat transfer coefficient. The total heat transfer resistance represented by ($U^{-1} = C_1 + C_2\, Re^{-n}$) in this analysis was patterned after Wilson (1915) that accommodated the complexities associated with the determination of the two-phase heat transfer coefficient and the buildup of surface scaling resistances. They presented the analysis of a water-cooled condenser and an air-cooled evaporator with supporting numerical examples that were based on the thermoeconomic optimization procedure of this study.

Bejan (1987) presented a review article to outline the most basic steps of the procedure of entropy generation minimization (thermodynamic design) at the system-component level. His current paper was a continuation of his earlier review work (Bejan, 1982a and 1982b). As a result, a further objective was to review the fundamental work published in this area in the 1980s. The researcher focused on the fundamental mechanisms responsible for the generation of entropy in heat and fluid flow and on the design tradeoff of balancing the heat transfer irreversibility against the fluid flow irreversibility. He selected applications from the fields of heat exchanger design, thermal energy storage, and mass exchanger design. This current article provided a comprehensive, up-to-date review of second-law analyses published in the heat and mass transfer literature during the last decade.

Bejan (1988) summarized the structure of heat exchanger irreversibility as follows:

$$N_s = N_{s,imbalance} + N_{s,\Delta T} + N_{s,\Delta P} \tag{30}$$

The first term on the right hand side represents the remanent (flow-imbalance) irreversibility. The second term on the right hand side represents the heat transfer irreversibility. The third term on the right hand side represents the fluid flow irreversibility. The researcher suggested to calculate the remanent irreversibility ($N_{s,imbalance}$) first in the thermodynamic optimization of any heat exchanger because it is not logic to invest heat exchanger area and "engineering" into minimizing the sum ($N_{s,\Delta T}$ + $N_{s,\Delta P}$) when this sum is already negligible compared with the remanent irreversibility ($N_{s,imbalance}$). Only in very special case does the entropy generation rate of a heat exchanger break into a sum of these three terms. One such case is the balanced counter flow heat exchanger in the nearly balanced and nearly ideal limit ($\omega \to 1$, $\Delta T \to 0$, ΔP's $\to 0$). This case was discussed in details in Bejan (1977).

The remanent (flow-imbalance) irreversibility of two-stream parallel-flow heat exchangers can be obtained by combining the equation of the entropy generation rate of the entire heat exchanger with the perfect design conditions and the effectiveness relation for parallel flow (Bejan (1993)) as follows:

$$N_{s,imbalance} = \frac{\dot{S}_{gen}}{(\dot{m}c_p)_2} = \ln\left\{\left(\frac{T_2}{T_1}\right)^{\omega}\left[1+\left(\frac{T_1}{T_2}-1\right)\frac{\omega}{1+\omega}\right]^{1+\omega}\right\} \tag{31}$$

In the limit of extreme imbalance ($\omega \to \infty$), Eq.(31) becomes

$$N_{s,imbalance} = \frac{T_2}{T_1} - 1 - \ln\frac{T_2}{T_1} \tag{32}$$

In this limit, the side 1 stream is so large that its temperature remains equal to T_1 from inlet to outlet. It behaves like a stream that condenses or evaporates at constant pressure.

On the other hand, the remanent (flow-imbalance) irreversibility of two-stream counter flow heat exchangers can be obtained as follows:

$$N_{s,imbalance} = \frac{\dot{S}_{gen}}{(\dot{m}c_p)_2} = \ln\left\{\left[1-\frac{1}{\omega}\left(1-\frac{T_2}{T_1}\right)\right]^{\omega}\frac{T_1}{T_2}\right\} \tag{33}$$

From Eq. (31) and Eq. (33), it is clear that the remanent (flow-imbalance) irreversibility in parallel flow is greater than in counterflow. Also, both flow arrangement approach the value indicated by Eq. (32) as the flow imbalanced ratio (ω) increases.

Sekulic and Milosevic (1988) investigated entropy generation in heat exchanger networks using the component balance approach.

Witte (1988) used the second-law efficiency to develop a new technique for optimizing the design of heat exchangers. His method related the operating costs of the exchanger to the destruction of availability caused by the exchanger operation. The researcher related directly the destruction of availability to the second-law efficiency of the exchanger. This allowed one to find the *NTU* at which the benefits of reduced availability losses were offset by the costs of added area; this was the optimal point. In order to determine the proper cost of irreversibility to be used in the optimization process, he included the irreversibility cost in a dimensionless parameter that represented the ratio of annual ownership costs to annual operating costs that included irreversibility costs. In this way, every heat exchanger designer could estimate the costs of irreversibilities for his particular system, and then used the generalized method that was developed herein for determining the optimal heat exchanger size. His method was applicable to any heat exchanger for which the ε-*NTU*-*R* relationships were known.

Grazzini and Gori (1988) developed a general expression for entropy generation in counter-current heat exchangers is. Their expression was applicable to incompressible liquids and perfect gases. They defined two new entropy generation numbers, N_M and N_Q.

They investigated the relative position of both the maximum and minimum in the entropy generation numbers. They applied their analysis to an air-air counter-current heat exchanger. The three entropy generation numbers, N_s, N_M and N_Q, had a different variation with *NTU* at the different values of the capacity flow rate ratio employed in the calculations.

Eğrican (1989) investigated logarithmic mean temperature difference (LMTD) method based on the first law of thermodynamics with effectiveness-transfer unit methods and entropy generation units based on the second law of thermodynamics. To give an example, the researcher applied this method to counter-flow shell and tube heat exchanger.

Poulikakos and Johnson (1989) obtained a general expression for the entropy generation for combined convective heat and mass transfer in external flows. This expression took into account irreversibilities due to the presence of heat transfer across a finite temperature difference, mass transfer across a finite difference in the chemical potential of a species, and flow friction. Minimizing the entropy generation in heat- and fluid-flow devices was a valuable criterion for optimum design. The researchers showed that the same philosophy could be used when in addition to heat transfer and fluid flow irreversibilities, mass transfer irreversibilities existed in the thermal system of interest. They applied the general expression for entropy generation to two fundamental problems of forced convection heat and mass transfer, namely, laminar and turbulent boundary layer forced convection from a flat plate and from a cylinder in crossflow. After minimizing the entropy generation, they drew useful conclusions that were representative of the second law viewpoint for the definition of the optimum operating conditions for the specified applications.

Paoletti et al. (1989) calculated the exergetic losses in compact heat exchanger passages. In their approach, the researchers analyzed the heat exchangers on the basis of second-law and used the entropy generation rate in a local sense. They associated the symbol *Be* as an alternative irreversibility distribution parameter and defined as the ratio of heat transfer irreversibility to total irreversibility due to heat transfer and fluid friction

$$Be = \frac{\dot{S}_{gen,\Delta T}}{\dot{S}_{gen,\Delta T} + \dot{S}_{gen,\Delta P}} = \frac{1}{1+\phi} \tag{34}$$

In addition, Benedetti and Sciubba (1993) called it the Bejan number (*Be*). Later, Natalini and Sciubba (1999) introduced also Bejan number (*Be*) using Eq. (34). Natalini and Sciubba (1999) solved first the full Navier-Stokes equations of motion for turbulent viscous flow, together with the appropriate energy equation, via a standard finite-element code with a k-epsilon closure, to obtain complete velocity and temperature fields. Then, the researchers used these fields to compute the entropy generation rates corresponding to the viscous and thermal dissipation.

It is clear from Eq. (34) that $Be = 1$ that occurs at $\phi = 0$ corresponds to the case at which the irreversibility is dominated by the heat transfer effects. On the other hand, $Be = 0$ corresponds to the case at which the irreversibility is dominated by the fluid friction effects. Also, $Be = 0.5$ that occurs at $\phi = 1$ corresponds to the case at which the heat transfer irreversibility and the fluid friction irreversibility are equal.

It should be noted that the *Be* definition in Eq. (34) should not be confused with another Bejan number (*Be*) used in convection. Petrescu (1994) defined the Bejan number (*Be*) as follows:

$$Be = \frac{\Delta P L^2}{\mu \alpha} \tag{35}$$

This was similar to the new dimensionless group developed by Bejan and Sciubba (1992) in their study on the optimal spacing between plates cooled by forced convection. Also, the same group appeared in the solutions to other electronic cooling problems involving forced convection (Bejan, 1993). In addition, the group defined in Eq. (35) governed all the phenomena of contact melting and lubrication, in both internal and external contact configurations (Bejan, 1992).

The researcher reported that the Bejan number (*Be*) was essential in at least four areas of heat transfer: electronic cooling, scale analysis of forced convection, second law analysis of heat exchangers, and contact melting and lubrication. The *Be* group defined by Eq. (35) was the forced convection ($Pr \geq 1$) analog of the Rayleigh number (*Ra*) for natural convection in $Pr \geq 1$ fluids.

Sekulic (1990) presented the entropy generation (irreversibility) concept as a convenient method for estimating the quality of the heat exchange process in heat exchanger analysis. The researcher used the entropy generation caused by finite temperature differences, scaled by the maximum possible entropy generation that could exist in an open system with two fluids, as the quantitative measure of the quality of energy transformation (the heat exchange process). This quality was defined as

$$\begin{aligned}&\text{Quality of energy transformation} = 1 - (\text{Entropy generation in the real process})\\&\times(\text{Entropy generation in the most disadvantageous case})^{-1}\end{aligned} \tag{36}$$

According to this concept, entropy generation = 0 (reversible process) corresponded to the highest quality, and the quality of energy transformation decreased with increasing entropy generation. Another point that should be considered was that the use of this concept required the determination of the most disadvantageous case. Substituting the entropy generation number (N_s) and the maximum possible dimensionless entropy generation ($N_{s,max}$) into Eq. (36) gives the following quantity termed as "Heat Exchange Reversibility Norm" (*HERN*):

$$Y_s = 1 - \frac{N_s}{N_{s,max}} \tag{37}$$

HERN is a measure of the quality of energy transformation of heat exchangers. In his analysis, it was assumed that the contribution of fluid friction to entropy generation was negligible. If the pressure drop contribution to the total irreversibility was not negligible, then, it must be taken into account. The quality of the heat transfer process in a heat exchanger was dependent on the following three quantities for this special case: the ratio of inlet temperatures, the ratio of heat capacity rates, and the effectiveness of the heat exchanger.

Sekulic (1990) applied the *HERN* measure to a two-fluid heat exchanger of arbitrary flow arrangement. He discussed the effect of various parameters (inlet temperature ratio, fluid flow heat capacity rate ratio, flow arrangements) and the heat exchanger thermal size (number of heat transfer units) on the quality of energy transformation for various types of heat exchangers.

Rispoli and Sciubba (1990) investigated numerically the calculation of local irreversibilities in compact heat exchangers. Their approach to analyze heat exchangers on the basis of second law was to use the entropy generation rate in a local sense. They analyzed two various geometries of compact heat exchanger passages on the basis of local entropy generation rate. The evaluation of the entropy production in a local sense had the following advantages:

i. it was possible to assess the effect of design changes both on the local and the global irreversibility,
ii. direct and consistent comparisons between various design configurations, both from the designer's and from the user's perspective, could be made,
iii. entropy productionmaps of different devices and/or components can be established, and the overall system design rationalized.

The coupled momentum and energy equations should be solved to determine the local entropy production rates. The corresponding entropy production was computed by using the resulting velocity and temperature fields.

The researchers defined local Bejan number as the ratio of entropy generation due to thermal effects to total entropy generation as:

$$Be = \frac{\text{Local entropy generation rate due to thermal effects}}{\text{Total local entropy generation rate}} = \frac{\dot{S}_t}{\dot{S}_t + \dot{S}_v} \tag{38}$$

Bejan number (Be) ≈ 1 for high Re flows, low Pr fluids, high logarithmic mean temperature difference ($LMTD$) while Bejan number (Be) ≈ 0 for low Re flows, high Pr fluids, low logarithmic mean temperature difference ($LMTD$). Theoretically, Bejan number (Be) = 0 only for totally isothermal flows.

Evans and von Spakovsky (1991) set forth two fundamental principles of differential Second Law analysis for heat exchanger design. Their first principle defined a Second Law temperature, while their second principle defined a Second Law temperature difference. The researcher showed that the square of the ratio of the Second Law temperature difference to the Second Law temperature was always to be equal to the negative of the partial derivative of the rate of entropy generation (for heat transfer) with respect to the overall conductance of the heat exchanger. For the basic design of elementary heat exchangers, every of these two Second Law quantities was shown to take the form of a simple geometric average.

Sieniutycz and Shiner (1994) presented a review article on thermodynamics of irreversible processes and its relation to chemical engineering: Second law analyses and finite time thermodynamics. In spite of their focus was on chemical engineering applications, their stated objective was to clarify the connections between the work of different groups in the field.

Later, Bejan (1996c) presented notes on the history of the method of entropy generation minimization (finite time thermodynamics). The researcher mentioned that Professors Sieniutycz and Shiner deserved credit for conducting a review of this wide and active field. Also, credit went to their calloborators, Professors Berry and Ratkje, who had clearly contributed to their review article. In these notes, he wanted to complement Sieniutycz and Shiner's list with a few additional references that shed a somewhat various light on the age and origins of the method. In brief, the method was older than portrayed in Sieniutycz and Shiner (1994) and its roots were in engineering, not in physics.

Bejan (1996b) presented a review article on entropy generation minimization: the new thermodynamics of finite-size devices and finite-time processes. His review traced the development and adoption of EGM method in many sectors of mainstream thermal engineering and science: cryogenics, heat transfer, education, storage systems, solar power plants, nuclear and fossil power plants, and refrigerators. The researcher placed emphasis on the fundamental and technological importance of the optimization method and its results, the pedagogical merits of the method, and the chronological development of the field.

Xiong et al. (1996) discussed some conceptual problems in their paper. Firstly, according to the physical meaning of effectiveness, the researchers developed a new expression of effectiveness using an ideal heat exchanger model and temperature histogram method, in which the non-uniform inlet temperature profile was considered. Secondly, they studied the relation of entropy generation number (N_s) to effectiveness (ε). They pointed out that both of them could express the perfect degree of a heat exchanger to the second thermodynamic law. Finally, they presented a criterion named as comprehensive thermal performance coefficient ($CTPE$) to describe both quantity and quality of heat transferred in a heat exchanger.

Xu et al. (1996) demonstrated the difference between the entropy generation number method proposed by Bejan and the method of entropy generation per unit amount of heat transferred in analyzing the thermodynamic performance of heat exchangers. The researchers pointed out the reason for leading to the above difference. They proposed a modified entropy generation number for evaluating the irreversibility of heat exchangers is which was in consistent with the entropy generation per unit amount of heat transferred in entropy generation analysis. Also, they investigated entropy generated by friction. Their results showed that when the entropy generated by friction in heat exchangers was taken into account, there was a minimum total entropy generation number while the *NTU* and the ratio of heat capacity rates varied. The existence of this minimum was the prerequisite of heat exchanger optimization.

Ogulata and Doba (1998) presented a cross-flow plate type heat exchanger that had been studied and manufactured in the laboratory conditions because of its effective use in waste heat recovery systems. The researchers tested this new heat exchanger with an applicable experimental set up, considering temperatures, velocity of the air and the pressure losses occurring in the system. They measured these variables and determined the efficiency of the system. They took into consideration the irreversibility of the heat exchanger while they performed the heat exchanger design so that the minimum entropy generation number had analyzed with respect to second law of thermodynamics in the cross-flow heat exchanger.

Ogulata et al. (1999) studied and manufactured a cross-flow plate-type heat exchanger in laboratory conditions because of its effective use in waste heat recovery systems. The researcher tested this new heat exchanger with an applicable experimental setup, considering temperatures, velocity of the air, and the pressure losses occurring in the system. They measured these variables and determined the efficiency of the system. The heat exchanger irreversibility was taken into consideration, while the heat exchanger design was such that the minimum entropy generation number was analyzed with respect to the second law of thermodynamics in the cross-flow heat exchanger. The minimum entropy generation number was dependent on the parameters called the optimum flow path length and dimensionless mass flux. They analyzed variations of the entropy generation number with these parameters.

Nafey (2000) presented theoretical analysis of entropy generation and availability destruction of *NTU* similar cocurrent or countercurrent heat exchangers connected in series. The researcher developed a criterion for comparing the relative performance of any number of in-series connected similar heat exchangers. He presented the effect of various influencing parameters like the number of connected heat exchangers, the individual effectiveness of every unit, the heat capacity rate ratio and flow arrangement on the quality of heat exchange. He found that the maximum of availability destruction (maximum entropy generation) for in-series-connected similar cocurrent heat exchangers was obtained at $\varepsilon^* = (1+R)^{-1}$. However, for counter-current heat exchangers connected in-series; $\varepsilon_n^* = (\sum_{i=0}^{n} R^{i/n})^{-1}$. This analysis might be useful for a proper choice of the number of heat exchangers to be connected together and the choice for the best operating conditions.

Ordóñez and Bejan (2000) determined the main architectural features of a counterflow heat exchanger based on thermodynamic optimization subject to volume constraint. The researchers assumed that the channels were formed by parallel plates, the two fluids were ideal gases, and the flow was fully developed, laminar or turbulent. First, they showed that the irreversibility of the heat exchanger core was minimized with respect to (1) the ratio of the two-channel spacings, and (2) the total heat transfer area between the two streams. Second, the entropy generation rate also accounted for the irreversibility due to discharging the spent hot stream into the ambient. They showed that the design could be optimized with respect to (1), (2) and (3) the ratio of the capacity rates of the two streams. The optimized features of the geometry were robust with respect to whether the external discharge irreversibility was included in the entropy generation rate calculation.

Hesselgreaves (2000) reviewed the different approaches to second law analysis and presented a rational method that satisfied the physical requirements. His intention was not reviewing all previous work, but presenting an approach that resolved some perceived inconsistencies and paradoxes. The researcher derived entropy generation numbers for different types of two-fluid heat exchangers with zero pressure drop and finite pressure drop. The types of heat exchangers were: heat exchangers with flow imbalance, unbalanced counterflow, parallel flow, condensing on one side, and evaporation on one side. An important result of this investigation was that the basic entropy generation relationship for gas flows was controlled by the flow Mach number. This was consistent with an extension of Shapiro (1953)'s classical one-dimensional flow analysis of a compressible gas with friction and heat addition.

Yilmaz et al. (2001) presented second-law based performance evaluation criteria to evaluate the performance of heat exchangers. First, the researchers recalled and discussed the need for the systematic design of heat exchangers using a second law-based procedure. Then, they classified the evaluation techniques for heat exchangers based on the second law of thermodynamics into two categories: the evaluation techniques using entropy as an evaluation parameter, and the evaluation techniques using exergy as an evaluation parameter. They presented and reviewed collectively both categories, and gave their respective characteristics and constraints. It was shown how some of these criteria were related to every other. Also, emphasis was placed on the importance of second law-based thermoeconomic analysis of heat exchangers, and these methods were discussed briefly.

Vargas et al. (2001) studied the process of determining the internal geometric configuration of a component by optimizing the global performance of the installation, which used the component. The example chosen was the crossflow heat exchanger used in the environmental control system of a modern aircraft. The researchers achieved the optimization of global performance by minimizing the total entropy generation rate of the installation. There were three degrees of freedom in the heat exchanger configuration (the length-to-width and height-to-width aspect ratios, and the separator plate spacing ratio) that was subjected to two global constraints: total component volume, and total wall material volume (or weight/density) of wall material. Their numerical results showed how the optimal configuration responded to changes in specified external parameters like volume, weight, Mach number, diffuser inlet cross-sectional area, and the pressure at which the cabin air was initially bled from the engine compressor. They showed that the optimal configuration was robust and that major features like the ratios of channel spacings and flow lengths were relatively insensitive to changes in some of the external parameters. Also, they showed that the optimal heat exchanger geometry was insensitive to the thermodynamic irreversibility caused by discharging the used ram air into the ambient.

Vargas and Bejan (2001) showed that the main geometric features of a flow component could be deduced from the thermodynamic optimization of the global performance of the largest flow system that incorporated the component. Their approach represented a departure from the usual approach, where a flow component was optimized in isolation. The researchers chose the counterflow heat exchanger of the environmental control system (*ECS*) used on modern aircraft as an example. They fitted the heat exchanger with a diffuser and a nozzle for the ram air, and the ECS run on the boot strap air cycle, employing an additional compressor and turbine. They considered two heat transfer surface types, finned and smooth parallel plates. They reported numerical results for the external geometric aspect ratios of the heat exchanger, and for the plate-to-plate spacing of the smooth-plates model. They showed that the optimized geometry for the core with finned surfaces was nearly the same as the optimized geometry for the core with smooth plates. Many optimized geometric features were robust with respect to changes in external parameters that varied from one application to the next. Their method illustrated in this work – the thermodynamic (constructal) optimization of flow geometry – was applicable to any system that run on the basis of a limited amount of fuel (exergy) installed onboard, e.g., automobiles, ships, portable tools.

Bejan and Lorente (2001) reviewed recent developments in thermodynamic optimization by focusing on the generation of optimal geometric form (shape, structure, topology) in flow systems. The flow configuration was free to vary. The researchers drew examples of large classes of applications from different sectors of mechanical and civil engineering: the distribution of heat transfer area in power plants, optimal sizing and shaping of flow channels and fins, optimal aspect ratios of heat exchanger core structures, aerodynamic and hydrodynamic shapes, tree-shaped assemblies of convective fins, tree-shaped networks for fluid flow and other currents, optimal configurations for streams that undergo bifurcation or pairing, insulated pipe networks for the distribution of hot water and exergy over a fixed territory, and distribution networks for virtually everything that moves in society (goods, currency, information). The principle-based generation of flow geometry united the thermodynamic optimization developments known in mechanical engineering with lesser known applications in civil engineering and social organization. Their review article extended thermodynamics, because it showed how thermodynamic principles of design optimization took into account the development of optimal configurations in civil engineering and social organization.

Bejan (2001) discussed the basis for the entropy generation minimization method, and a series of key applications in power generation, refrigeration, and exergy conservation. The researcher started with a review of the concept of irreversibility, entropy generation, or exergy destruction. He used the proportionality between exergy destruction and entropy generation in the search for improved thermodynamic performance subject to finite-size constraints and specified environmental conditions. He gave examples from refrigeration, energy storage systems for sensible heat and latent heat, solar energy, and the generation of maximum power by using a stream of hot gas. He showed that the physical structure (geometric configuration, topology) of the system springed out of the process of global thermodynamic optimization subject to global constraints. This principle generated structure not only in engineering but also in physics and biology (constructal theory).

Shiba and Bejan (2001) showed that the internal geometric configuration of a component could be deduced by optimizing the global performance of the installation, which used the component. The example chosen was the counterflow heat exchanger, which served as condenser in a vapor-compression-cycle refrigeration system for environmental control of aircraft. The researcher achieved the optimization of global performance by minimizing the total power requirement or the total entropy generation rate. There were three degrees of freedom in the heat exchanger configuration that was subjected to two global constraints: total volume, and total volume (or weight) of wall-material. Their numerical results showed how the optimal configuration responded to changes in specified external parameters like refrigeration load, fan efficiency, and volume and weight. In accordance with constructal theory and design (Bejan, 2000), it was shown that the optimal configuration was robust: major features like the ratio of diameters and the flow length were relatively insensitive to changes in the external parameters.

Bejan (2002) discussed the fundamentals of the methods of exergy analysis and entropy generation minimization (or thermodynamic optimization-the minimization of exergy destruction). The researcher began with a review of the irreversibility concept, entropy generation, or exergy destruction. Examples illustrated the accounting for exergy flows and accumulation in closed systems, open systems, heat transfer processes, and power and refrigeration plants. He gave examples from energy storage systems for sensible heat and latent heat, solar energy, and the generation of maximum power in a power plant model with finite heat transfer surface inventory.

Yuan and Kou (2001) investigated the entropy generation in a crossflow heat exchanger including three gas streams and the influence of longitudinal wall conduction on the entropy generation. Using the numerical method, the researchers calculated the exit mean temperature of every stream, and then computed the number of entropy generation units. Their results indicated that the entropy generation increased with the decrease of inlet temperature of gas stream 3 and the decrease of inlet temperature ratio of gas streams 1 to 2. Also, their results showed that the longitudinal wall conduction raised the entropy generation and that this raising increased with increasing NTU when heat capacity rate ratio of stream 1 was 0.5

Yuan and Kou (2003) investigated the entropy generation on a crossflow heat exchanger including three gas streams with three various arrangements. Using the numerical method, the researchers calculated individually the exit mean temperature of every gas stream in various arrangements, and then computed the number of entropy-generation units of every arrangement. Their results indicated that there was a maximum entropy generation for every arrangement along with the increase in number of transfer units (*NTU*). Comparing the three arrangements showed that the entropy generation of the third arrangement was the lowest, because this arrangement transferred heat across a smaller temperature difference. Also, this study examined the influence of longitudinal wall conduction on the entropy generation in every arrangement. The largest influence on entropy generation was found in the third arrangement.

Shah and Skiepko (2004) found that the concept of minimum irreversibility was not quite applicable to the heat exchanger analysis although it was associated with the maximum energy efficiency for energy conversion processes in thermal systems. The researchers showed that the heat exchanger effectiveness could be maximum, having an intermediate value or minimum at the maximum irreversibility operating point depending on the flow arrangement of the two fluids. Similarly, the heat exchanger effectiveness could be minimum or maximum at the minimum irreversibility operating point. They illustrated and discussed such heat exchanger performance and irreversibility trends by combining the temperature difference irreversibility with the *P-NTU* results for complex flow arrangements.

Strub et al. (2004) evaluated the contribution of second law analysis to the study of a phase changing of ice slurries as secondary refrigerant in cooling systems. First, the researchers calculated the enthalpies and the entropies. Then, they carried out an entropy/exergy analysis of a heat exchanger. Their work attempted to provide a thermodynamic criterion to choose the kind of fluid and the inflow conditions that were more suitable for a particular application. They established the method and obtained the results for an ethyl alcohol-water mixture.

Lerou et al. (2005) studied optimization of counterflow heat exchanger (*CFHX*) geometry through minimization of entropy generation. In their study, the researchers applied another, less familiar design strategy where different loss mechanisms such as pressure drop and parasitic heat flows were all treated as a production of entropy. Thus, it was possible to compare and sum them. In this way, they found that a *CFHX* configuration was optimal for a certain application, producing a minimum of entropy and thus had minimum losses. For instance, they gave the design steps of a *CFHX* for the micro cooling project at the University of Twente. Also, they presented a generalization of micro *CFHX* dimensions for cooling powers between 10 and 120 mW.

Mohamed (2006) realized analysis of heat transfer and fluid flow thermodynamic irreversibilities on an example of a counter flow double pipe heat exchanger utilizing turbulent air flow as a working fluid. During the process of mathematical model creation and for various working and constructing limitations, the researcher studied total thermodynamic irreversibility. His work proved that the irreversibility occurred due to unequal capacity flow rates (flow imbalance irreversibility). He concluded that the heat exchanger should be operated at effectiveness, $\varepsilon > 0.5$ and the well operating conditions would be achieved when $\varepsilon \sim 1$ where low irreversibility was expected. He adopted a new equation to express the entropy generation numbers for imbalanced heat exchangers of similar design with smallest deviation from the exact value. His new equation was yhe sum of two terms: the first term was the contribution of the pressure terms and the second term was the contribution of the temperature terms. He compared the results obtained from his new equation with the exact values and with those obtained by Bejan (1988). Also, the guide charts presented in his work could be used to determine the most wanted combination of the effects of various parameters to obtain minimal irreversibility.

Naphon (2006) presented the theoretical and experimental results of the second law analysis on the heat transfer and flow of a horizontal concentric tube heat exchanger. The researcher designed and constructed the experiments setup for the measured data. He used hot water and cold water as working fluids. He did the test runs at hot and cold water mass flow rates in the range of 0.02-0.20 kg/s. The inlet hot water and inlet cold water temperatures were between 40 and 50 °C, and between 15 and 20 °C, respectively. He discussed the influences of the inlet conditions of both working fluids flowing through the heat exchanger on the heat transfer characteristics, entropy generation, and exergy loss. Based on the conservation equations of energy, he developed his mathematical model and solved using the central finite difference method to obtain temperature distribution, entropy generation, and exergy loss. The predicted results obtained from the model were validated by comparing with his measured data. From this comparison, he found that there was reasonable agreement between predicted results and those from the measured data.

Kurtbaş et al. (2007) investigated the effects of propeller-type turbulators located in the inner pipe of co-axial heat exchanger on entropy generation rate (N_s) and exergy loss rate (E^*). Propeller-type turbulators with blade angles (θ) = 10°, 20° and 40°, also at every angle the propellers with blade diameter (d_b) = 48 mm, 50 mm and 52 mm. The researchers mounted these turbulators in the inner pipe using different distances (L_t). According to the flow observation experiments, they found maximum decaying distance of swirl flow as 30 cm. They performed the experiments with various distances of turbulators. In this system, they investigated heat transfer, entropy generation rate, and exergy loss rate. Then, they investigated the influences of angle, diameter, and number of the blades on the heat transfer, entropy generation rate, and exergy loss rate and compared with every other for various values of the Reynolds number, from 104 to 3×10^4, and Prandtl number equal to 0.71. They found that Nusselt number and exergy loss rate approximately increased from 95 to 354 and 0.04 to 0.2 depending on blade angle, interturbulator distance and propeller diameter for $104 < Re < 3\times10^4$. The heat exchanger efficiency changed at between 0.17 to 0.72 levels.

Khan et al. (2007) were specifically interested in determining an optimal design of the tube banks in cross flow using an entropy generation minimization method that was as a unique measure to study the thermodynamic losses caused by heat transfer and pressure drop for a fluid in cross flow with tube banks. The optimal design of tube banks was very important because of extensive use of high performance compact heat exchangers that were found in many applications like an automobile radiator, an oil cooler, a preheater, an air-cooled steam condenser, a shell and tube type heat exchanger, and the evaporator of an air conditioning system. Usually, tube banks were arranged in an inline or staggered manner, where one fluid moved across the tubes, and the other fluid at a different temperature passed through the tubes. In their study, both inline and staggered arrangements were studied and their relative performance was compared for the same thermal and hydraulic conditions. The researchers employed the crossflow correlations for the heat transfer and pressure drop to calculate entropy generation rate. They obtained a general dimensionless expression for the entropy generation rate by considering a control volume around a tube bank and applying conservation equations for mass and energy with entropy balance. Analytical/empirical correlations for heat transfer coefficients and friction factors were used, where the characteristic length was used as the diameter of the tubes and reference velocity used in Reynolds number and pressure drop was based on the minimum free area available for the fluid flow. Also, they performed a parametric study to show the influences of various design variables on the overall performance of tube banks. They showed that all relevant design parameters for tube banks, including geometric parameters and flow conditions, could be simultaneously optimized.

Gupta and Das (2007) carried out the second law analysis of crossflow heat exchangers in the presence of non-uniformity of flow. The researchers modeled this non-uniformity with the help of axial dispersion model and took into account the back mixing and flow maldistribution. They evaluated an analytical model for exergy destruction for the cross-flow configuration. They carried out a wide range of study of the operating parameters and non-uniform flow on exergetic behavior of crossflow heat exchangers. Their results clearly brought out not only the reason behind the maximum entropy paradox in heat exchangers but also the proper perspective of exergy destruction and the consequent optimization of crossflow heat exchangers from the second law viewpoint.

Gupta et al. (2007) studied second law analysis of counter flow cryogenic heat exchangers in presence of ambient heat-in-leak and longitudinal conduction through wall. The researchers carried out this study because the performance of highly effective heat exchangers was strongly dependent on these irreversibilities in low temperature applications. They observed that the influence of ambient heat-in-leak was different for the balanced and imbalanced counter flow high *NTU* heat exchangers. Also, their study made it possible to compare the different irreversibilities for varying range of *NTU* and analyze the effect of external irreversibilities on the performance of heat exchangers when either hot fluid or cold fluid was minimum capacity fluid.

Pitchandi and Natarajan (2008) described the second law of thermodynamics analysis of a regenerative heat exchanger. Their analysis was based on the fact that the dimensionless parameters, known as the reduced periods and reduced length, were the characteristic variables to describe the heat exchanger. The researcher discretized the solid matrix in the heat exchanger passage using trapezoidal rule and took the elemental matrix as a thermodynamic system. They applied the second law of thermodynamics to the system and obtained the entropy generation equation using the dimensionless numbers Reduced period (Π) and Reduced length (Λ) in every element. They studied the variation of entropy generation due to reduced length and reduced period. Also, the effect of the effectiveness of the heat exchanger on entropy generation was highlighted.

He et al. (2009) applied second-law based thermodynamics analysis to a new heat exchanger with helical baffles. The helical baffles were designed as quadrant ellipses and every baffle occupied one quadrant of the cross-section of the shell side. The researchers carried out experimental tests with cold water in the tube side with a constant flow rate, and hot oil on the shell side with flow rate range from 4–24 m^3/h. They measured the temperatures and pressures for the inlet and outlet of both sides. They investigated heat transfer, pressure drop, entropy generation, and exergy loss of the new heat exchanger and compared with the results for a conventional shell-and-tube heat exchanger with segmental baffles. The computed results indicated that both the entropy generation number and exergy losses of the new heat exchanger design were lower than those of the heat exchanger with segmental baffles that meaned that the novel heat exchanger had a higher efficiency than the heat exchanger with segmental baffles, from the second-law based thermodynamics viewpoint.

Fan and Luo (2009) presented the experimental results of second law analysis on the heat transfer and hydraulic characteristics of a mini crossflow heat exchanger equipped with constructal distributor/collector. In their experiments, the researchers used hot and cold water as working fluids. They performed tests for different "distributor-heat-exchanger-collector" configurations at channel Reynolds numbers in the heat exchanger between 800 and 3100. They discussed the integration of constructal component on the thermal performance, entropy generation, exergy loss, and the second law effectiveness of the heat exchanger. Also, they analyzed and discussed the relationship between heat-transfer intensification and system-irreversibility production in this case.

Guo et al. (2009) developed a new shell-and-tube heat exchanger optimization design approach using entropy generation minimization and genetic algorithm. The researchers employed the dimensionless entropy generation rate obtained by scaling the entropy generation on the ratio of the heat transfer rate to the inlet temperature of cold fluid as the objective function. They took some geometrical parameters of the shell-and-tube heat exchanger as the design variables and applied the genetic algorithm to solve the associated optimization problem. They showed that for the case that the heat duty was given, not only could the optimization design increased the heat exchanger effectiveness significantly, but also decreased the pumping power dramatically. In the case that the heat transfer area was fixed, the benefit from the increase of the heat exchanger effectiveness was much more than the increasing cost of the pumping power.

Exergy change rate in an ideal gas flow or an incompressible flow can be divided into two types: a thermal exergy change rate and a mechanical exergy loss rate. San (2010) generalized the mechanical exergy loss rates in the two flows using a pressure-drop factor ($F_{\Delta P}$) because the consumed mechanical exergy is usually more valuable than the recovered thermal exergy for heat exchangers using in waste heat recovery. The researcher proposed a weighing factor to modify the pressure-drop factor. He defined an exergy recovery index (η_{II}) and expressed it as a function of effectiveness (ε), ratio of modified heat capacity rates (C^*), hot stream-to-dead-state temperature ratio (T_h/T_o), cold stream-to-dead-state temperature ratio (T_c/T_o) and modified overall pressure-drop factor ($F^*_{\Delta P}$). This η_{II}-ε relation could be used to find the η_{II} value of a heat exchanger with any flow arrangement. He established the η_{II}-NTU and η_{II}-NTU_h relations of cross-flow heat exchanger with both fluids unmixed respectively. The former provided a minimum NTU design principle and the latter provided a minimum NTU_h design principle. A numerical example showed that, at a fixed heat capacity rate of the hot stream, $(\dot{m}\, c_p)_h$, the heat exchanger size yielded by the minimum NTU_h principle was smaller than that yielded by the minimum NTU principle.

Guo et al. (2010) presented a multi-objective optimization of heat exchanger thermal design in the framework of the entropy generation minimization. Their objectives were to minimize the dimensionless entropy generation rates related to the heat conduction under finite temperature difference and fluid friction under finite pressure drop. The researchers specified constraints using the admissible pressure drop and design standards. They employed the genetic algorithm to search the Pareto optimal set of the multi-objective optimization problem. They found that the solutions in the Pareto optimal set were trade-off between the pumping power and heat exchanger effectiveness. The optimal solution in the Pareto optimal set achieved the largest exchanger effectiveness by consuming the least pumping power under the design requirements and standards. In comparison with the single-objective optimization design, the multi-objective optimization design led to the significant decrease in the pumping power for achieving the same heat exchanger effectiveness and presented more flexibility in the design process.

Kotcioglu et al. (2010) studied a second law analysis of a cross-flow heat exchanger (*HX*) in the presence of a balance between the entropy generation due to heat transfer and fluid friction. The researchers investigated the entropy generation in a cross-flow *HX* with a new winglet-type convergent-divergent longitudinal vortex generator (*CDLVG*). They presented optimization of *HX* channel geometry and effect of design parameters regarding the overall system performance. Based on the entropy generation minimization (*EGM*), they developed the optimization model for the *HX* flow lengths and *CDLVGs*. They found that increasing the cross-flow fluid velocity enhanced the heat transfer rate and reduced the heat transfer irreversibility. Their test results demonstrated that the *CDLVGs* were potential candidate procedure to improve the disorderly mixing in channel flows of the cross-flow type *HX* for large values of the Reynolds number.

Wang et al. (2010) studied experimentally flow and heat transfer characteristics of the shell-and-tube heat exchanger (*STHXs*) with continuous helical baffles (*CH-STHX*) and segmental baffles (*SG-STHX*). In their experiments, these *STHXs* shared the same tube bundle, shell geometrical structures, different baffle arrangement, and number of heat exchange tubes. Their experimental results suggested that the *CH-STHX* could increase the heat transfer rate by 7-12% than the *SG-STHX* for the same mass flow rate although its effective heat transfer area had 4% decrease. Also, the heat transfer coefficient and pressure drop of the *CH-STHX* had 43-53% and 64-72% increase than those of the *SG-STHX*, respectively. Based on second-law thermodynamic comparisons in which the quality of energy were evaluated by the entropy generation number and exergy losses, the *CH-STHX* decreased the entropy generation number and exergy losses by 30% and 68% on average than the *SG-STHX* for the same Reynolds number. Also, the analysis from nondimensional correlations for Nusselt number and friction factor revealed that if the maximal velocity ratio $R > 2.4$, the heat transfer coefficient of *CH-STHX* was higher than that of *SG-STHX*, and the corresponding friction factor ratio kept at constant $f_{0,CH}/f_{0,SG} = 0.28$.

Assad (2010) presented a theoretical analysis of a heat exchanger with a negligible fluid flow pressure drop to determine whether it was better to operate the heat exchanger with the minimum or maximum heat capacity rate of the hot fluid from entropy generation point of view. The researcher derived entropy generation numbers (N_s) for both cases, and his results showed that they were identical, when the heat exchanger was running at a heat capacity ratio (R) = 0.5 with heat exchanger effectiveness (ε) = 1. He defined an entropy generation number ratio (S^*) by dividing the entropy generation number for minimum heat capacity rate on the hot fluid side to the entropy generation number for maximum heat capacity rate on the hot fluid side. S^* had a maximum value at $\varepsilon = (1+R)^{-1}$ for any inlet temperature ratio (T_r) and R values. This result could be obtained by taking the derivative of S^* with respect to ε and equating it to zero. When R = 0.1, 0.5 and 0.9, the entropy generation number ratio (S^*) received a maximum value at an effectiveness (ε) = 0.91, 0.67 and 0.526, respectively. When R = 0.9, the entropy generation number ratio (S^*) was the same for all inlet temperature ratios (T_r) at $\varepsilon = 0.8$. However, when $\varepsilon < 0.8$, S^* increased as T_r decreased, and when $\varepsilon > 0.8$, S^* increased as T_r increased. His results showed that the entropy generation number ratio (S^*) was far from 1 depending on the inlet temperature ratio (T_r) of the cold and hot fluid. When $S^* < 1$, it was better to run the heat exchanger with minimum heat capacity rate on the hot fluid side, whereas when $S^* > 1$, it was better to run the heat exchanger with maximum heat capacity rate on the hot fluid side. These results could be used to determine the wanted combination of the effects of various parameters (R, T_r, and ε) to obtain lower irreversibility. Also, these results were valid for parallel-flow and counterflow heat exchangers.

Fakheri (2010) further explored the topic of an ideal heat exchanger that was still an open question. It was shown that the minimization of entropy production or exergy destruction should not be an objective in heat exchanger design. It was further proven that heat exchanger effectiveness did not correlate with irreversibility. Therefore, the researcher introduced a new performance measure to characterize the performance of heat exchangers, entropy flux (Γ), which was allowing the comparison of different heat exchangers under varying operating conditions by applying the second law. The entropy flux (Γ) could be defined as

$$\Gamma = \frac{\dot{S}}{UA} \tag{39}$$

As shown in Eq. (39), entropy flux (Γ) incorporated three main features of heat exchangers, namely, entropy generated ($\dot{S}$) that so far was only a result of heat transfer, overall heat transfer coefficient (U) and the heat exchanger area (A). In heat exchanger design, the goal is to increase the heat transfer while reducing the size so that higher values of entropy flux are desirable. For a given effectiveness, a single stream heat exchanger had the absolute maximum entropy flux, and for capacity ratios (C_r) greater than zero, counterflow had the highest entropy flux, parallel flow the lowest, and the shell and tube heat exchangers were somewhere in between.

On the basis of the first and second laws of thermodynamics, Ruan et al. (2011) derived the general expression of the number of entropy generation units of three-fluid heat exchangers with three thermal communications. The researchers discussed thoroughly the effect of several non-dimensional design parameters on the number of entropy generation units of three-fluid heat exchangers. Furthermore, they gave the detailed comparisons of results for the arrangement of the parallel flow and the counter flow. They showed that the variation tendencies of the number of entropy generation units with the ratio of the thermal resistances, ratio of the thermal capacities, and number of heat transfer units for the parallel-flow arrangement were different from those of the counter-flow arrangement. There was an extremum of the number of entropy generation units for the counter-flow arrangement. Also, the entropy generation for the counter flow was mostly smaller than that of the parallel flow under the same conditions

Arivazhagan and Lokeswaran (2011) investigated the entropy generation rate in shell and tube heat exchanger with porous medium inserted inside the tubes. The researchers used three various waste metal chips made of copper, aluminum, and mild steel as porous medium. There was a trade-off between the pressure drop and heat transfer in the design of enhanced heat exchangers. If Reynolds number increased, the rate of heat transfer would also increase at the expense of reasonable pressure drop In porous flow. Because of turbulent energy dissipation at high Reynolds number, this pressure drop would increase further, resulting in high entropy generation. They developed and used the empirical correlations for the entropy generation minimization of the actual heat exchanger. They derived their conclusions on the basis of the behavior of the entropy generation number (N_s) as a function of the Reynolds number (Re). On the basis of the entropy generation minimization, they found the upper limit of Reynolds number to be 1450, beyond which irreversibility increased.

1.2 Optimization of internal enhancements

In many heat transfer applications, internal enhancements are utilized to promote or enhance heat transfer. However, any enhancement of a primary surface gives rise to an increase in pressure drop for a given mass flow rate. Using thermodynamic optimization, we may also assess the penalties of improving thermal contact in terms of entropy generation. Since thermal enhancement leads to higher heat transfer rates, a lower mass flow is permissable in most applications. Therefore, in a given application with fixed duty (Q), the temperature difference will be reduced for the same mass flow rate. This tradeoff, potentially allows for the overall entropy generation rate to be reduced below that for the primary surface at the desired duty condition. In this section, we will examine the impact of thermal enhancement devices such as strip fins and ribs on entropy generation.

Bejan and Pfister (1980) used entropy generation as a measure of the relative merit of heat transfer augmentation techniques relative to each other and to the heat exchange apparatus in which they might be incorporated. In this way, heat transfer augmentation techniques were viewed as design changes capable of reducing the irreversible destruction of useful energy (exergy) in heat exchange equipment. The entropy generation rate took into account simultaneously the heat transfer and fluid friction changes associated with implementing a heat transfer augmentation technique. The researchers proposed that the merit of a given heat transfer augmentation technique might be evaluated by comparing the rate of entropy generation of the heat exchange apparatus before and after the implementation of the augmentation technique. Using in-tube roughness as an instance, they showed what specific operating conditions must be met before the destruction of exergy could be reduced via heat transfer augmentation.

Benedetti and Sciubba (1993) presented a novel method that could be helpful in assessing the 'optimal' configuration of finned-tube heat exchangers. Their method was based on the determination on a local basis of the two components of the entropy generation rate: the one caused by viscous dissipations and the one due to thermal irreversibilities. Depending on the engineering purpose for which a technical device was designed, it could be argued that the 'optimal' configuration would be that in which either one (or both) of these two entropy generation rates was minimized. For a heat exchanging device, it was important to minimize thermal irreversibilities, but more important was to minimize the mechanical power lost in achieving a prescribed heat-exchange performance: to this purpose, one could form a 'relative irreversibility index' (named 'Bejan number (Be)' here and use it to assess the merit of a given configuration. In the procedure developed here, the researchers considered a circular, single-tube, finned heat exchanger configuration. They computed the velocity and temperature fields via a standard finite-element package (*FIDAP*) for a realistic value of the Reynolds number and for a variety of geometric configurations (different fin external diameters and fin spacing). Then, they calculated the entropy generation rate from the flow field, and examined both at a local level, to detect possible 'bad' design spots (i.e., locations that corresponded to abnormally high entropy generation rates that could be cured by design improvements), and at an 'overall' (integral) level, to assess the 'entropic' performance of the heat exchanger. They given 'Optimal' curves, and determined the 'optimal' spacing of fins using alternatively the entropy generation rate and the total heat transfer rate as objective functions: different optima arise, and the differences as well as the similarities were discussed in detail.

Another widely used thermal enhancement device used in heat transfer applications is the offset strip fin. Manglik and Fang (1994) applied the second law of thermodynamics to evaluate the heat transfer enhancement of offset strip fin core relative to plain plate fin compact heat exchangers. The researchers considered single-phase air flow in both laminar and turbulent regimes. examined entropy generation rates using the procedure proposed by Bejan and Pfister (1980). They presented the thermal-hydraulic performance in terms of area goodness factor (*j/f*) and the entropy generation number ($N_{s,a}$). Due to the irreversibility reduction was a trade-off between the heat transfer enhancement and the corresponding pressure drop penalty, they introduced a new parameter, entropy generation distribution factor (ψ) as

$$\psi = \frac{\dot{S}_{\Delta T,o} - \dot{S}_{\Delta T,a}}{\dot{S}_{\Delta P,a} - \dot{S}_{\Delta P,o}} \tag{40}$$

This new parameter represents the ratio of entropy generation reduction due to heat transfer enhancement and the increase in entropy generation due to the consequent increase in fluid friction. Thermodynamics benefits would be obtained only if $\psi > 1$. The magnitude of ψ was such that a better resolution was obtained for the entropy generation change due to the variations in operating conditions. They reported entropy generation numbers for three types of flow: constant mass flow rate, constant pressure drop, and constant pumping power. They delineated the relative effect of the aspect ratio, fin density, and fin thickness to offset length ratio of the offset strip fins on heat transfer enhancement and entropy generation minimization.

Sciubba (1996) presented a novel method that could be helpful in assessing the optimal configuration of finned-tube heat exchangers. His method was an extension of the local irreversibilities method, and it was based on the determination on a local basis of the two components of the entropy generation rate: the one caused by viscous dissipations and the one due to thermal irreversibilities. Depending on the engineering purpose for which a technical device was designed, it could be argued that the optimal configuration would be that in which either one (or both) of these two entropy generation rates was minimized. For a heat exchanging device, it was important to minimize thermal irreversibilities, but more important was to minimize the mechanical power lost in achieving a prescribed heat-exchange performance: to this purpose, one could form a relative irreversibility index (named Bejan number (*Be*) here), and use it to assess the merit of a given configuration. Average or global Bejan number (Be_{av}) could be found by integration of Eq. (38) as:

$$Be_{av} = \frac{\int_v \dot{S}_t}{\int_v \dot{S}_t + \int_v \dot{S}_v} = \frac{1}{1 + Br} \tag{41}$$

$$Br = Ec.\mathrm{Pr} = \frac{\mu V^2}{k \Delta T} \tag{42}$$

Average Bejan number (Be_{av}) $\rightarrow 0$ in the limit of vanishing average thermal gradient ΔT, and $Be_{av} \rightarrow 1$ in the limit of vanishing mean velocity gradient.

In the procedure presented here, the researcher considered a circular, single-tube, finned heat exchanger configuration. He computed the velocity and temperature fields (via a standard finite-element package, *FIDAP*) for a realistic value of the Reynolds number and for different geometric configurations (different fin external diameters and fin spacing). Then, he calculated the entropy generation rate from the flowfield, and examined both at a local level, to detect possible bad design spots (i.e., locations that corresponded to abnormally high entropy generation rates that could be cured by design improvements), and at an overall (integral) level, to assess the entropic performance of the heat exchanger. He gave optimal curves and determined the optimal spacing of fins using alternatively the entropy generation rate and the total heat transfer rate as objective functions: different optima arise, and the differences as well as the similarities were discussed in detail.

Tagliafico and Tanda (1996) presented a thermodynamic method for the comparison of plate fin heat exchanger performance. The researchers evaluated and scaled the entropy production of a given heat transfer surface geometry using that of corresponding reference configuration (a parallel-plate channel) with the same frontal area, volume, heat transfer duty, and mass flow rate to relate the relative merit of the surface geometry to corresponding irreversibility level. They applied their method to a number of plate-fin compact heat exchanger surfaces whose performance data were taken from Kays and London (1984). They examined six types of heat exchanger enhancements: the plain fin, louvered fin, strip fin, wavy fin, pin fin, and perforated fin. From this analysis, they found that the thermodynamic performance of the most suitable surfaces, among those considered in this study, turned out to be strongly related to the operating conditions (both heat transfer duty and mass flow rate). Also, they found that the strip fin was the thermodynamically most efficient augmentation device.

Muley and Maglik (1999) investigated performance optimization of plate heat exchangers with chevron plates. In this study, the researchers repeated Manglik and Fang's (1994) analysis but for corrugated rib surfaces used in plate heat exchangers. These devices also know as chevron ribs are widely used in process heat exchangers due there ease of construction and cleaning for fouling applications. They showed results for constant mass flow, constant pumping power, and constant pressure drop. They found that corrugated ribs at the fixed pumping power and fixed pressure drop constraints, led to a thermodynamically more efficient system.

Su et al. (1999) found a new way of fin design to minimize the irreversibilities due to heat transfer and fluid friction and maximize the available work of the working fluid. First, the researchers derived the general entropy generation formulas for fins according to the first and second law of thermodynamics. Then, they made a theoretical analysis on cylindrical pin fins and rectangular straight fins using the above formulas. They obtained the minimum entropy generation formulas for these two types of fins and proposed a principle for fin optimization, where the minimum entropy generation was chosen to be the objective function to be studied. They discussed in detail the influence of various parameters on fin entropy generation in forced convection heat transfer.

2. External structure

The ability of a designer to minimize the thermal resistance between the source of heat dissipation and the thermal sink is essential in controlling maximum operating temperatures. While the convective heat transfer coefficient could potentially be enhanced with an increase in the approach velocity, the dependence of heat transfer coefficient on the square root of the velocity in laminar flow results in diminished returns as velocity is increased. The second option for reducing film resistance is achieved by increasing the effective surface area for convective heat transfer. This is typically achieved through the use of heat sinks in single fluid heat exchangers and extended surfaces in two fluid heat exchangers. Heat sinks offer a low cost, convenient method for lowering the film resistance and in turn maintaining junction operating temperatures at a safe level in electronic components. Unfortunately, the selection of the most appropriate heat sink for a particular application can be very difficult given the many design options available. Thermal analysis tools, ranging from simple empirically derived correlations to powerful numerical simulation tools, can be used to analyze the thermal performance of heat sinks for a given set of design conditions. Regardless of which procedure is used, analysis tools only provide a performance assessment for a prescribed design where all design conditions are specified a priori. Following an exhaustive parametric analysis, design options can be assessed with respect to their influence on thermal performance, however, there is no guarantee that an "optimized" solution is obtained since the parametric analysis only provides a ranking of a limited set of test cases. The method of entropy generation minimization, pioneered by Bejan, provides a procedure for simultaneously assessing the parametric relevance of system parameters as they relate to not only thermal performance but also viscous effects.

2.1 Fin shape

Heat exchanger fins are often used in heat exchange devices to increase the heat transfer rate between the heat-exchange surface and the surrounding fluid. Extended surfaces (fins) enhance heat transfer rate by increasing surface area and by inducing turbulent mixing of flow. They can be found in many engineering applications such as the cooling of turbine blades in gas turbine engines, the cooling of electronic components, and different other heat exchange devices used in aerospace, aircraft, chemical processing plants, …, etc. There are different kinds of heat exchanger fins, ranging from relatively simple shapes, like rectangular, cylindrical, annular, tapered or pin fins, to a combination of various geometries. These fins may protrude from either a cylindrical or rectangular base.

Numerous analysis tools are available for determining the thermal performance of heat sinks given a well defined set of design conditions. Convective optimizations are available, such as those presented in Kraus and Bar-Cohen (1995), however, these models assumes a prescribed heat transfer coefficient over the length of the fins which is constant, while in most heat sink applications, hydrodynamic and thermal entrance effects introduce a variable heat transfer coefficient, at least over a portion of the heat sink. The assumption of a constant value of heat transfer coefficient can no longer be prescribed, since the value will depend upon fin spacing and length in the direction of flow. Optimization routines that lead to changes in fin spacing, fin height or fin length also result in changes in the mean heat transfer coefficient and head loss in such a way that iterative procedures are required. While in some instances parametric studies can be undertaken to obtain a relationship between thermal performance and design parameters, a comprehensive design tool should also take into consideration the effect of viscous dissipation and its relationship on thermal performance. The entropy generation associated with heat transfer and frictional effects serve as a direct measure of lost potential for work or in the case of a heat sinks and other finned systems. A modeling approach that establishes a relationship between entropy generation and a fin design parameters, can be used in such a manner that all relevant design conditions combine to produce the best possible thermal sink for the given constraints.

Poulikakos and Bejan (1982) established a theoretical framework for the minimization of entropy generation in forced convection for the design of extended surfaces by the use of the first and second laws of thermodynamics. First, the researchers derived the entropy generation rate formula for a general fin. The entropy generation rate for extended surfaces in external flow with conductive resistance was defined by the following relationship:

$$\dot{S}_{gen} = \frac{\dot{Q}\theta_b}{T_\infty^2} + \frac{F_d V_f}{T_\infty} \tag{43}$$

The temperature excess of the fin or heat sink (θ_b) might be related to the overall system thermal resistance:

$$\theta_b = \dot{Q} R_{fin} \tag{44}$$

Based on this general result, they developed analytical methods and graphic results for selecting the optimum dimensions of pin fins, rectangular plate fins, plate fins with trapezoidal cross section, and triangular plate fins with rectangular cross section.

Lee and Lin (1995) examined the performance and the entropy generation rate of a fractal-like fin under crossflow. This fin type was defined as a fin with subfins repeatedly extending in a fixed way.

Khan et al. (2006) examined the role of cross-sectional shape on entropy generation for several widely used fin cross-sections. The cross-sections examined were circular, elliptical, square, and rectangular. The researchers obtained a general dimensionless expression for the entropy generation rate by considering a control volume around the pin fin including base plate and applying the conservations equations for mass and energy with the entropy balance. They developed the formulation for the dimensionless entropy generation rate in terms of dimensionless variables, including the aspect ratio, Reynolds number, Nusselt number, and the drag coefficient. They examined selected fin geometries for the heat transfer, fluid friction, and the minimum entropy generation rate corresponding to various parameters including axis ratio, aspect ratio, and Reynolds number. Their results clearly indicated that the preferred fin profile was very dependent on these parameters. As the fin became more slender two effects contribute to the reduction in entropy generation number, namely increased surface area that reduced the temperature excess, and a reduction in profile drag which in turn reduced the viscous losses.

2.2 Plate fin arrays

It is well known that in plate fin type heat exchangers the backmixing and other deviations from plug flow contribute significantly to the inefficiency of the heat exchanger that is important to heat exchangers working in the cryogenic regime.

Culham and Muzychka (2001) presented a procedure that allowed the simultaneous optimization of heat sink design parameters for electronic applications based on a minimization of the entropy generation associated with heat transfer and fluid friction. All relevant design parameters for plate fin heat sinks, including geometric parameters, heat dissipation, material properties and flow conditions could be simultaneously optimized to characterize a heat sink that minimized entropy generation and in turn results in a minimum operating temperature. The researchers modified Eq. (43) to account for the overall sink resistance rather than the resistance of a single fin using a simple control volume analysis as follows:

$$\dot{S}_{gen} = \frac{\dot{Q}^2 R_{\sin k}}{T_\infty^2} + \frac{F_d V_f}{T_\infty} \tag{45}$$

Using Eq. (45), along with the appropriate expressions for the fin resistance, convective heat transfer coefficient, and frictional/drag losses, a model for the entropy generation rate was developed for an array of parallel plates.

Also, they integrated a novel approach for incorporating forced convection through the specification of a fan curve into the optimization procedure, providing a link between optimized design parameters and the system operating point. They presented examples that demonstrated the robust nature of the model for conditions typically found in electronic applications. It was not unusual for a designer to be given an overall maximum heat sink volume. The examples presented in Culham and Muzychka (2001) were assumed to be constrained by a overall maximum volume of 50 mm × 50 mm × 25 mm. In addition, it was assumed that a total heat dissipation of 30 W was uniformly applied over the base plate of the heat sink that had a uniform thickness of 2 mm. Other constraints that were fixed were the thermal conductivity of the heat sink at k = 200 W/m.K and the ambient temperature of the surrounding air medium at T_o = 25 °C or 298 K.

Culham and Muzychka (2001) presented several cases that demonstrated the method of entropy generation minimization for sizing plate fin heat sinks. Their examples included single and multi-parameter optimizations. Their results demonstrated the influence of introducing progressively more unconstrained variables into the optimization procedure. The system of non-linear equations for several cases could be solved using numerical procedures like Newton-Raphson solution, contained within many commercially available algebraic software tools. Given the geometric constraints and a uniform heat load to the base plate of the heat sink of 30 W, an optimum number of fins, N, was to be determined when V_f = 2 m/s, t = 1 mm, and H = 25 mm. As shown in Table 1, the estimation of the appropriate number of fins was $N \approx 29$. It was easily seen that decreasing the number of fins led to an increase in the thermal resistance of the heat sink which in turn led to an increase in the temperature excess and a resultant increase in the entropy generation rate. Increasing the number of fins beyond the optimized value would lead to a decrease in the heat sink resistance and temperature excess, but the increase in the head loss associated with fluid drag would result in an increase in the entropy generation rate.

While the optimization procedure estimated the optimum number of fins to be 28.57 the relatively wide range of near minimum entropy generation rate between $20 < N < 35$, provided designers with a range of options when specifying the appropriate number of fins. In subsequent applications of the optimization method, additional design variables were introduced into the procedure to simultaneously consider multiple parameters that led to an optimization of the temperature excess and the head loss of the heat sink.

Additional parameters were left unconstrained, like velocity (V_f), fin height (H), number of fins (N), and fin thickness (t). Case (ii) examined the influence of relaxing the constraint on free stream velocity prescribed in Case (i) while all other assumed constraints remained unchanged. As shown in Table 1, the optimized number of fins was determined to be $N \approx 27$ and the approach velocity was estimated to be V_f = 2.81 m/s for minimum entropy generation. A decrease in the number of fins and an increase in the free stream velocity led to a heat sink with a lower temperature excess but a higher head loss. Overall, the entropy generation rate for this case was lower than in the previous example. Case (iii) examined a three parameter optimization where the constraint on the fin thickness was removed. The results of the optimization gave $N \approx 38$, V_f = 3.28 m/s, and t = 0.4 mm as shown in Table 1. Further gains had been made in lowering the heat sink temperature excess and head loss that resulted in yet a further decrease in the entropy generation rate. However, the fin thickness might be too thin for practical manufacturing considerations. Finally, none of the variables of interest would be constrained to predetermined values, thus providing a simultaneous optimization of all design variables, including the free stream velocity (V_f), the number of fins (N), the fin thickness (t), and the fin height (H). Their results of the optimization gave $N \approx 19$, V_f = 1.21 m/s, t = 1.6 mm, and H = 122 mm. Once again a more optimal solution had been found. While the approach presented provided an optimized heat sink, the fin height exceeded the maximum allowable height of 25 mm predicated by the board-to-board spacing.

Moreover, it was important to note, that as more variables became unconstrained, the system was progressively seeking a more optimal design. For instance, in cases (ii) and (iii), although the fin count increased, the fin thickness decreased, leading not only to a thermally more efficient design, but also a system that used less material. Finally, one might introduce additional constraints as needed that limited the temperature excess or the mass of the heat sink. Their method outlined was also applicable to fin arrays used in heat exchangers.

Case	N	θ_b (°C)	ΔP (mmH$_2$O)	V_f (m/s)	t (mm)	H (mm)	$\dot{S}_{gen}$ (W/°C)
(i)	28.57	11.51	5.62	2.0	1.0	25	0.00435
(ii)	26.77	9.49	7.02	2.81	1.0	25	0.00402
(iii)	38.14	8.66	5.78	3.28	0.4	25	0.00370
(iv)	19.07	7.20	1.90	1.21	1.6	122	0.00290

Table 1. Optimized Conditions for All Test Cases.

Their model was shown to converge to a unique solution that gave the optimized design conditions for the imposed problem constraints.

The specification and design of heat sinks for electronic applications is not easily accomplished through the use of conventional thermal analysis tools because "optimized" geometric and boundary conditions are not known a priori.

Culham et al. (2007) presented an analytical model for calculating the best possible design parameters for plate fin heat sinks using an entropy generation minimization procedure with constrained variable optimization. The researchers adapted the method to include a thermal spreading resistance in the overall thermal circuit. Their method characterized the contribution to entropy production of all relevant thermal resistances in the path between source and sink as well as the contribution to viscous dissipation associated with fluid flow at the boundaries of the heat sink. The minimization procedure provided a fast, convenient method for establishing the "best case" design characteristics of plate fin heat sinks given a set of prescribed boundary conditions. They showed that heat sinks made of composite materials containing nonmetallic constituents, with a thermal conductivity as much as an order of magnitude less that typical metallic heat sinks, could provide an effective alternative where performance, cost, and manufacturability were of importance. Also, they showed that the spreading resistance encountered when heat flows from a heat source to the base plate of a heat sink, while significant, could be compensated for by making appropriate design modifications to the heat sink.

Iyengar and Bar-Cohen (2003) presented a coefficient of performance (*COPT*) analysis for plate fin heat sinks in forced convection and showed to provide a viable technique for combining least-material optimization with the entropy minimization methodology. The *COPT* metric related the heat sink cooling capability to the invested fan pumping work and the thermodynamic work required to manufacture and assemble the heat sink. The proposed optimization methodology maximized the forced convection cooling that could be achieved by a heat sink occupying a specified volume, with a fixed energy investment and entropy generation rate. Also, their study identified the presence of an optimal resource allocation ratio, providing the most favorable distribution of existing energy resources, between heat sink manufacturing and operation, over a fixed product life cycle.

Abbassi (2007) investigated the entropy generation in a uniformly heated microchannel heat sink (*MCHS*). He used analytical approach to solve forced convection problem across *MCHS*. This analytical approach was a porous medium model based on extended Darcy equation for fluid flow and two-equation model for heat transfer. Simultaneously, closed form velocity solution in a rectangular channel was employed to capture z-directional viscous effect diffusion and its pronounced influence on entropy generation through fluid flow. Subsequently, governing equations were cast into dimensionless form and solved analytically. Then, second law analysis of problem was conducted on the basis of obtained velocity and temperature fields and expressions for local and average entropy generation rate were derived in dimensionless form. Then, average entropy generation rate was utilized as a criterion for assessing the system performance. At the end, the effect of influential parameters like, channel aspect ratio (α_S), group parameter (Br/Ω), thermal conductivity ratio (C) and porosity (ε) on thermal and total entropy generation was investigated. In order to examine the accuracy of the analysis, the results of thermal evaluation were compared to one of the previous investigations conducted for thermal optimization of MCHS.

Khan et al. (2009) employed an entropy generation minimization (*EGM*) procedure to optimize the overall performance of microchannel heat sinks. The researchers developed new general expressions for the entropy generation rate by considering an appropriate control volume and applying mass, energy, and entropy balances. They investigated the influence of channel aspect ratio, fin spacing ratio, heat sink material, Knudsen numbers, and accommodation coefficients on the entropy generation rate in the slip flow region. They used analytical/empirical correlations for heat transfer and friction coefficients, where the characteristic length was used as the hydraulic diameter of the channel. In addition, a parametric study was performed to show the effects of various design variables on the overall performance of microchannel heat sinks.

The thermal design of plate fin heat sinks can benefit from optimization procedures where all design variables are simultaneously prescribed, ensuring the best thermodynamic and air flow characteristic possible. While a cursory review of the thermal network established between heat sources and sinks in typical plate fin heat sinks would indicate that the film resistance at the fluid-solid boundary dominates, it is shown that the effects of other resistance elements, such as the spreading resistance and the material resistance, although of lesser magnitude, play an important role in the optimization and selection of heat sink design conditions.

Zhou et al. (2009) proposed the multi-parameter constrained optimization procedure integrating the design of experiments (*DOE*), response surface models (*RSM*), genetic algorithm (*GA*), mixed integer optimization (*MOST*), and computational fluid dynamics (*CFD*) to design the plate finned heat sinks by minimizing their rates of entropy generation. The results of three cases demonstrated that the combination optimization algorithm was feasible. In these cases, the overall rate of entropy generation decreased as the result of introducing the additional constrained variables into the optimization procedure. As a result, the general thermal and fluid performance of the heat sink was dramatically improved.

Based on the results derived by the optimization, the researchers investigated the overall thermal and fluid performance of the plate finned heat sinks with both side and top bypass flow. Also, they established two correlations describing Nusselt number and friction factor, as the functions of geometrical and operational parameters, by means of the multivariate non-linear regression analysis. They deduced the specific expressions to compute the thermal resistance and the rate of entropy generation.

Ganzarollia, and Altemania (2010) performed the thermal design of a counterflow heat exchanger using air as the working fluid with two distinct goals: minimum inlet temperature difference and minimum number of entropy generation units. The researchers constituted the heat exchanger by a double-finned conductive plate closed by adiabatic walls at the fin tips on both sides. The cold and hot air flows were considered in the turbulent regime, driven by a constant pressure head. The thermal load was constant, and an optimization was performed in order to obtain the optimum fin spacing and thickness, according to the two design criteria. They employed a computer program to evaluate the optimum conditions based on correlations from the literature. They compared the results obtained from both design criteria to each other. They performed a scale analysis considering the first design goal and compared the corresponding dimensionless parameters with the results from the correlations.

Zhang et al. (2010) developed a general three-dimensional distributed parameter model (*DPM*) for designing the plate-fin heat exchanger (*PFHE*). The proposed model that allowed for the varying local fluid thermophysical properties inside the flow path could be applied for both dry and wet working conditions by using the uniform enthalpy equations. The researchers generated the grids in the *DPM* to match closely the flow passage of the heat exchanger. They adopted the classical correlations of the heat transfer and the flow friction to avoid solving the differential equations. As a result, the computation burden of *DPM* became significantly less than that of the Computational Fluid Dynamics method. They performed the optimal design of a *PFHE* based on the *DPM* with the entropy generation minimization taken into consideration. They employed the genetic algorithm to conduct the optimization due to its robustness in dealing with complicated problems. The fin type and fin geometry were selected optimally from a customized fin database. The *PFHE* included in an environmental control system was designed by using the proposed approach in their study. Finally, They evaluated the cooling performance of the optimal *PFHE* under both dry and wet conditions.

Galvis and Culham (2010) used the entropy generation minimization (*EGM*) method to find the optimum channel dimensions in micro heat exchangers with a uniform heat flux. With this approach, pressure drop and heat transfer in the micro channels were considered simultaneously during the optimization analysis. The researchers developed a computational model to find the optimum channel depth knowing other channel geometry dimensions and coolant inlet properties. Their assumptions were laminar and both hydrodynamically and thermally fully developed flow, and incompressible. However, they introduced the Hagenbach factor (*K*) to take into account the developing length effect in the friction losses. The Hagenbach factor (*K*) for rectangular channels obtained by Steinke and Kandlikar (2006) as follows:

$$\mathrm{K} = 0.6796 + 1.2197\alpha_S + 3.3089\alpha_s^2 - 9.5921\alpha_s^3 + 8.9089\alpha_s^4 + 2.9959\alpha_s^5 \qquad (46)$$

The micro channels were assumed to have an isothermal or isoflux boundary condition, non-slip flow, and fluid properties had dependency on temperature accordingly. For these particular case studies, the pressure drop and heat transfer coefficient for the isothermal boundary condition is lower than the isoflux case. As the channel size decreased, they found higher heat transfer coefficient and pressure drop. The optimum channel geometry that minimized the entropy generation rate tended to be a deep, narrow channel.

Rao and Patel (2010) discussed the use of particle swarm optimization (*PSO*) algorithm for thermodynamic optimization of a cross flow plate-fin heat exchanger. The researchers considered minimization of total number of entropy generation units for specific heat duty requirement under given space restrictions, minimization of total volume, and minimization of total annual cost as objective functions and treated individually. Based on the applications, they considered heat exchanger length, fin frequency, numbers of fin layers, lance length of fin, fin height and fin thickness or various flow length of the heat exchanger for optimization. They included heat duty requirement constraint in the procedure. Also, they presented two application examples to demonstrate the effectiveness and accuracy of the proposed algorithm. They validated the results of optimization using *PSO* by comparing with those obtained by using genetic algorithm (*GA*). In addition, they carried out parametric analysis to demonstrate the influence of heat exchanger dimensions on the optimum solution. Moreover, they presented the influence of variation of *PSO* parameters on convergence and optimum value of the objective.

Ahmadi et al. (2011) conducted a thermal modeling for optimal design of compact heat exchangers to minimize cost and entropy generation. The researchers applied an ε - *NTU* method for estimation of the heat exchanger pressure drop, and effectiveness. Fin pitch, fin height, fin offset length, cold stream flow length, no-flow length, and hot stream flow length were considered as six decision variables. They applied fast and elitist nondominated sorting genetic algorithm (i.e., nondominated sorting genetic algorithm II) to minimize the entropy generation units and the total annual cost (sum of initial investment and operating and maintenance costs) simultaneously. The results for Pareto-optimal front clearly revealed the conflict between two objective functions, the number of entropy generation units (N_s) and the total annual cost (C_{total}). It revealed that any geometrical changes that decreased the number of entropy generation units, led to an increase in the total annual cost and vice versa. Moreover, they derived an equation for the number of entropy generation units versus the total annual cost for the Pareto curve for prediction of the optimal design of the plate fin heat exchanger as follows:

$$C_{\text{total}}\ (\$) = \frac{-2.819N_s^3 - 4.311N_s^2 + 1.728N_s - 0.04891}{N_s^2 + 21.84N_s - 1.867} \times 10{,}000 \quad 0.0939 < N_s < 0.13 \quad (47)$$

Considering a numerical value for the number of entropy generation units in the range $0.0939 < N_s < 0.13$ provided the minimum total annual cost for that optimal point along with other optimal design parameters. Also, optimization of heat exchangers based on considering exergy destruction revealed that irreversibilities, like pressure drop and high temperature difference between cold and hot streams, played a key issue in exergy destruction. Thus, more efficient heat exchanger led to have a heat exchanger with higher total cost rate. At the end, the sensitivity analysis of change in the optimum number of entropy generation units and the total annual cost with change in the decision variables of the plate fin heat exchanger was also performed, and the results were reported.

Shuja and Zubair (2011) presented a detailed second-law based thermoeconomic optimization for a finned heat sink array. This involved including costs associated with material and irreversible losses due to heat transfer and pressure drop. The researchers optimized the effect of important physical, geometrical and unit cost parameters on the overall finned array for some typical operating conditions that were representative of electronic cooling applications. They presented the cost optimized results in terms of different parameters for a finned system. Furthermore, they explained the methodology of obtaining optimum design parameters for a finned heat sink system that would result in minimum total cost.

Gielen et al. (2011) discussed the use of second law based cost functions in plate fin heat sink design. The researchers proposed and compared a new entropy-based cost function with existing heat sink cost functions. A case study of a plate fin heat sink pointed out that their newly developed cost function offered a heat sink that was more than twice as efficient as a heat sink designed with the traditional thermal resistance minimization objective. The influences of this new heat sink design on data center cooling systems were considered and found to be significantly improving the system efficiency and waste heat recovery.

Al-Obaidi (2011) used second law analysis for a steady-state cross flow microchannel heat exchanger (*MCHX*) because this type of heat exchangers was known for its higher heat transfer coefficient and higher area per volume ratio. As a result, broad range studies were being carried out to optimize its performance and minimize its inefficiencies. The researcher employed entropy generation and exergy loss to investigate a multiport serpentine slab *MCHX* with ethylene glycol-water and air as the working fluids. She used conservation of energy and the increase in entropy principles to create a mathematical model that used various like heat capacity rate ratio, fluids inlet temperatures, effectiveness and pressure drop for obtaining entropy generation. Results were found on the basis of the behavior of the entropy generation number (N_s) with the key parameters. She found a good agreement between the predicted and the measured results.

2.3 Pin fins

For heat transfer enhancement, pin fins are widely used as effective elements. For this purpose, extensive work is being carried out to choose and optimize pin fins for different applications. Any optimization procedure would lead to desirable results only if the parallel pressure drop and heat transfer are considered. Pin fin arrays are another popular geometry used in electronics cooling. Pin fins are attractive as a result of their ability to operate easily in multi-directional fluid streams.

First, Lin and Lee (1997) conducted the second law analysis on a pin-fin array under crossflow to evaluate the entropy generation rate. Increasing the crossflow fluid velocity enhancing the heat transfer rate and hence, reducing the heat transfer irreversibility. Nevertheless, owing to the simultaneous increase in drag force exerting on the fin bodies, the hydrodynamic irreversibility increased also. An optimal Reynolds number thereby existed over wide operating conditions. The researchers searched the optimal design/operational conditions on the basis of entropy generation minimization. Also, they made the comparison of the staggered and the in-line pin-fm alignments.

Şara et al. (2001) presented heat transfer and friction characteristics, and the second law analysis of the convective heat transfer through a rectangular channel with square cross-sectional pin fins attached over a fiat surface. The researchers used different clearance ratios and interfin distance ratios and determined optimum pin-fin arrays that minimized entropy generation. They found that average Nusselt number based on the projected area decreased with increasing clearance ratio and interfin distance ratio, whereas average Nusselt number based on the total heat transfer area increased with increasing interfin distance ratio and with decreasing clearance ratio. Also, they found that the friction factor to decrease with increasing clearance ratio and interfin distance ratio. They obtained smaller entropy generation numbers at lower Reynolds number, higher clearance ratio, and higher interfin spacing ratio.

Khan et al. (2005) applied an entropy generation minimization (EGM) technique as a unique measure to study the thermodynamic losses caused by heat transfer and pressure drop in cylindrical pin-fin heat sinks. The researchers obtained a general expression for the entropy generation rate by considering the whole heat sink as a control volume and applying the conservation equations for mass and energy with the entropy balance. They used analytical/empirical correlations for heat transfer coefficients and friction factors in the optimization model, where the characteristic length was used as the diameter of the pin and reference velocity used in Reynolds number and pressure drop was based on the minimum free area available for the fluid flow. They studied both in-line and staggered arrangements and compared their relative performance on the basis of equal overall volume of heat sinks.

The details of the necessary models for heat transfer and drag might be found in Khan et al. (2005) along with the general control volume analysis. It was shown that all relevant design parameters for pin-fin heat sinks, including geometric parameters, material properties and flow conditions could be simultaneously optimized.

Khan et al. (2008) applied an entropy generation minimization (*EGM*) method to study the thermodynamic losses caused by heat transfer and pressure drop for the fluid in a cylindrical pin-fin heat sink and bypass flow regions. The researchers obtained a general expression for the entropy generation rate by considering control volumes around the heat sink and bypass regions. The conservation equations for mass and energy with the entropy balance were applied in both regions. Inside the heat sink, analytical/empirical correlations were used for heat transfer coefficients and friction factors, where the reference velocity used in the Reynolds number and the pressure drop was based on the minimum free area available for the fluid flow. In bypass regions theoretical models, based on laws of conservation of mass, momentum, and energy, were used to predict flow velocity and pressure drop. They studied both in-line and staggered arrangements and compared their relative performance to the same thermal and hydraulic conditions. Also, they performed a parametric study to show the effects of bypass on the overall performance of heat sinks.

Sahiti et al. (2008) derived experimentally the heat transfer and pressure drop characteristics of a double-pipe pin fin heat exchanger. The researchers used the empirical correlations previously validated with their experimental data to develop a mathematical model for the optimization of the actual heat exchanger. They developed the optimization model on the basis of the entropy generation minimization for various heat exchanger flow lengths and various pin lengths. They derived the conclusions on the basis of the behavior of the entropy generation number (N_s) as a function of the Reynolds number (*Re*). They showed that not all definition forms for the entropy generation number led to the right conclusions.

Nwachukwu and Onyegegbu (2009) derived an expression for the optimum pin fin dimension on exergy basis for a high temperature exchanger employing pin fins. Their result was different from that obtained by Poulikakos and Bejan (1982) for a low temperature heat recovery application. In addition, the researchers established a simple relation between the amounts the base temperature of the optimized pin fin was raised for a range of absorptive coating values. Employing this relation, if the absorptivity of the coating, the plate emissivity, the number of protruding fins, and some area and fluid parameters were known, they obtained immediately the corresponding value for the base temperature of the fin. Their analysis showed that the thermal performance of the exchanger improved substantially with a high absorptivity coating hence could be seen as a heat transfer enhancement feature of the exchanger operating with radiation dominance.

Kamali et al. (2010) used numerical analysis to investigate entropy generation for array of pin-fin heat sink. Technique was applied to study the thermodynamic losses caused by heat transfer and pressure drop in pin-fin heat sinks. The researchers obtained a general expression for the entropy generation rate by considering the whole heat sink as a control volume and applying the conservation equations for mass and energy with the entropy balance. They used analytical and empirical correlations for heat transfer coefficients and friction factors in the numerical modeling. Also, they studied heat transfer and pressure drop effects in entropy generation in control volume over pin-fins. They used numerical analysis for three different models of pin-fin heat sinks. The models were different in cross section area. These cross section areas were circle, horizontal ellipse, and vertical ellipse. Reference velocity used in Reynolds number and pressure drop was based on the minimum free area available for the fluid flow. As expected, the pressure drop and entropy generation increased with the rise of frontal velocity. Also, they investigated in-line arrangement of fins for numerical analysis and compared their relative performance. Finally, they compared the performance of these three models from the views point of heat transfer and total entropy generation rate. The elliptic pin fin showed the lowest pressure drops. Whereas, the circular geometry appeared as the best from the view point of the total entropy generation rate for low approach velocities and the elliptical geometry was the next favorable geometry from the view point of total entropy generation rate for higher approach velocities. Eventually, vertical elliptic fins showed the highest pressure drop and entropy generation among these different geometries.

Su et al. (2011) studied theoretically the entropy generation during heat transfer of a pin fin array in channels with lateral ejection holes for a turbine blade. The researchers established the entropy generation model based on the second-law analysis. They analyzed the distribution of the entropy generation due to heat transfer and fluid friction irreversibility respectively and made a comparison for three typical short pin fin channels. The entropy generation number component due to heat transfer decreased while Re_d increased, while the component due to fluid friction increased with the increase of Re_d. The entropy generation number (N_s) reached minimum when the two components met and the corresponding Reynolds number (Re_d) was optimal. The ejection holes affected the energy lost of the working fluid. For the three cases studied in this work, case b with short ejection holes gave the best comprehensive thermal performance with comparison to cases with no and long ejection holes. They suggested that their results would be helpful for the design of the heat dissipation of pin fin heat exchangers.

3. Conclusion

This chapter provides a comprehensive, up-to-date review in a chronological order on the research progress made on entropy generation minimization (thermodynamic optimization, or finite time thermodynamics). *EGM* is the method which combines into simple models the most basic concepts of heat transfer, fluid mechanics, and thermodynamics (Bejan, 1982a). These simple models are used in the real (irreversible) devices and processes optimization, subject to finite-size and finite-time constraints. The current review is related to using *EGM* method in heat exchangers for both internal structure and external structure. Examples are drawn from different kinds of applications: parallel flow, counterflow, crossflow, phase-change heat exchanger optimizations, as well as optimization of internal enhancement. Attention is also gives to micro systems such as microchannel heat exchanger (*MCHX*).

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5. Nomenclature

A	total heat transfer area, m^2
B	duty parameter
Be	Bejan number
Br	Brinkman number = $Ec.Pr$
C	thermal conductivity ratio
C^*	modified ratio of heat capacity rates, $(\dot{m}\, c_p)_h/(\dot{m}\, c_p)_c$
C_r	capacity ratios
C_{total}	total annual cost, $
c_p	constant-pressure specific heat, J/kg.K
d	diameter, m
E^*	exergy loss rate
Ec	Eckert number
F_d	total drag force on the fin (or array), N
$F_{\Delta P}$	pressure-drop factor
$F^*_{\Delta P}$	modified pressure-drop factor
f	Fanning friction factor
G	mass flux, $kg/m^2.s$
H	height, m
h	heat transfer coefficient, $W/m^2.K$, enthalpy, J/kg
j	Colburn factor
K	Hagenbach factor
k	thermal conductivity, W/m.K
L	length, m
$\dot{m}$	mass flow rate, kg/s
N	number of fins
N_s, N_M, N_Q	entropy generation numbers
Nu	Nusselt number
NTU	number of heat transfer units, $U_oA_o/(\dot{m}\, c_p)_{min}$
NTU_h	modified number of heat transfer units, $U_oA_o/(\dot{m}\, c_p)_h$
P	temperature effectiveness for a fluid, pressure, Pa
p	perimeter, m
Pe	Peclet number = $Re.Pr$
Pr	Prandtl number = ν/α
$\dot{Q}$	heat dissipation rate, W
dq/dx	heat transfer rate per unit length, W/m
R	heat capacity ratio
R_{fin}	resistance of the fin structure as a function of geometry, K/W
R_{sink}	overall resistance for the sink array, K/W
Ra	Rayleigh number
Re	Reynolds number
$\dot{S}_t$	entropy generation rate due to thermal effects, W/K

$\dot{S}_v$	entropy generation rate due to viscous dissipation, W/K
S^*	entropy generation number ratio
St	Stanton number
T	temperature, K
T_o	ambient temperature or dead-state temperature, K
T_r	inlet temperature ratio
T_∞	environment temperature, K
t	fin thickness, m
U	overall heat transfer coefficient, W/m^2.K
V	velocity, m/s
V_f	free stream or approach velocity, m/s
v	specific volume, m^3/kg
w_C	channel width, m
Y_s	heat exchange reversibility norm (HERN)

Greek

α	thermal diffusivity = $k/\rho c_p$, m^2/s
α_S	channel aspect ratio = H/w_C
Δ	difference
ε	effectiveness, porosity
ϕ	irreversibility distribution ratio
Γ	entropy flux
η	efficiency
η_{II}	exergy recovery index
Λ	Reduced length
μ	dynamic viscosity, kg/m.s
ν	kinematic viscosity, m^2/s
Π	Reduced period
θ	blade angle
θ_b	temperature excess of the fin, $(T_b - T_\infty)$
ρ	density, kg/m^3
Ω	dimensionless temperature difference
ω	capacity ratios
ψ	entropy generation distribution factor

Subscripts

0	without augmentation
1	stream 1
2	stream 2
av	average
b	blade
c	cold stream
gen	generation
h	hydraulic, hot stream
in	in
max	maximum value

min minimum value
o dead state or external (air side)
opt optium
out out
W-S Witte–Shamsundar

Superscripts

* at maximum irreversibility

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Selecting and bringing together matter provided by specialists, this project offers comprehensive information on particular cases of heat exchangers. The selection was guided by actual and future demands of applied research and industry, mainly focusing on the efficient use and conversion energy in changing environment. Beside the questions of thermodynamic basics, the book addresses several important issues, such as conceptions, design, operations, fouling and cleaning of heat exchangers. It includes also storage of thermal energy and geothermal energy use, directly or by application of heat pumps. The contributions are thematically grouped in sections and the content of each section is introduced by summarising the main objectives of the encompassed chapters. The book is not necessarily intended to be an elementary source of the knowledge in the area it covers, but rather a mentor while pursuing detailed solutions of specific technical problems which face engineers and technicians engaged in research and development in the fields of heat transfer and heat exchangers.

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